AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 47 No. 10 OCTOBER 1957 PRICE: 3s. 6d.

Rotary filing by air tools with carbide tips

does a better job for less money

A DIE SHOP that used to finish stub axle dies with hammer, chisel and grinder, cut the job time from 50 hours per die to 35 by switching to Atlas Copco rotary filing tools with carbide tips.

A truck manufacturer used *one* of these Atlas Copco carbide tips to finish 400 special truck frames—a job that had previously required 350 mounted points.

Atlas Copco air tools get high efficiency from carbide tips through vibrationless operation. The driving spindle runs in ample-size SKF ball bearings. Components are precision-made. The collet chuck holds the cutter shank dead centre, even at highest working speeds. The tough vane and turbine motors aren't affected by overloading.

The present range of Atlas Copco filing machines meets all needs for handy and highly efficient tools to take advantage of the potentialities of tungsten carbide cutters.

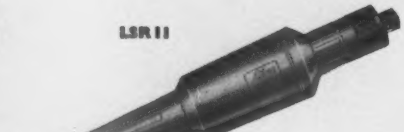
World-wide sales and service

The Atlas Copco Group embraces Atlas Copco companies or agents manufacturing or selling and servicing Atlas Copco equipment in ninety countries throughout the world. For further details of the equipment featured here, contact your local Atlas Copco Company or Agent. If you have any difficulty please write to Atlas Copco AB, Stockholm 1, Sweden, or Atlas Copco (Great Britain) Limited, Beresford Avenue, Wembley, Middlesex.

Trimming an impeller with a TS2F6H cutter. Tungsten carbide cutter heads saved about 40% on costs in this operation, compared to mounted points.

RAB 2FI

LSR 11



TS2F 6H

RAB 2FI cutter takes large cutters. Useful for single handed work in close quarters. 15,000 r.p.m.

LSR 11 for $\frac{3}{8}$"—$\frac{5}{8}$" heads. Convenient twist throttle. Built-in lubricator and silencer. 21,000 r.p.m.

TS2F6H for smaller $\frac{1}{4}$" heads. Precision-made components ensure vibrationless running 65,000 r.p.m.

Atlas Copco *Manufacturers of Stationary and Portable Compressors, Rock-Drilling Equipment, Loaders, Pneumatic Tools and Paint-Spraying Equipment.*

YI.

CERAMIC CUTTING TOOLS

The wide publicity given to cemented oxide cutting tools during recent months has promoted the belief in some quarters that a new metal-cutting medium of astronomical potential is now available.

Although such tools were demonstrated by Wickman Limited at the International Machine Tools Exhibition in 1956, after development work extending over many years, it has always been realised that severe limitations to further research and development are imposed by the lack of machine tools capable of extending their known high-speed machining characteristics to the ultimate.

Experience with cemented oxide tools already gained suggests that where very high speeds can be applied with relatively light cutting pressures, satisfactory performances can be achieved sometimes comparable with those being regularly obtained with cemented carbides.

In an endeavour to discover the natural field of application for cemented oxides and the techniques of tool geometry, speeds and feeds, and the potential metal removal ratings, this Company has now begun a new programme of tests. New machine tools of advanced experimental design have been specially built and are now being installed for this purpose. We think these tests will provide conclusive evidence of the fields of application of most economic benefit and we shall look forward to keeping industry informed of their progress.

In the meantime, we are prepared to supply cemented oxide tips in a limited range of shapes to cutting tool users who may wish to test for themselves the possibilities of the material, and who may possess suitable machines for this purpose.

WICKMAN LIMITED.

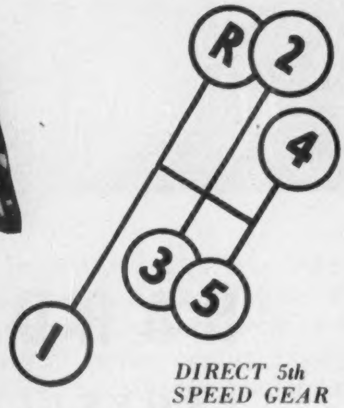
488 ST

PROVED IN SERVICE

In the medium vehicle range, the David Brown Model 542 5-speed gearbox is acknowledged as the outstanding unit, and its unfailing reliability, over very high mileages, has earned it a world-wide reputation.

Offered with either centre or forward control and a choice of direct or over-drive top gear, the 542 gearbox is designed for a maximum torque of 205 lb/ft. and is characterised by its sturdiness, compactness, low weight to strength ratio, ease and quietness of operation.

Write for publications E393.26 and 27



DIRECT 5th SPEED GEAR

THE

DAVID BROWN

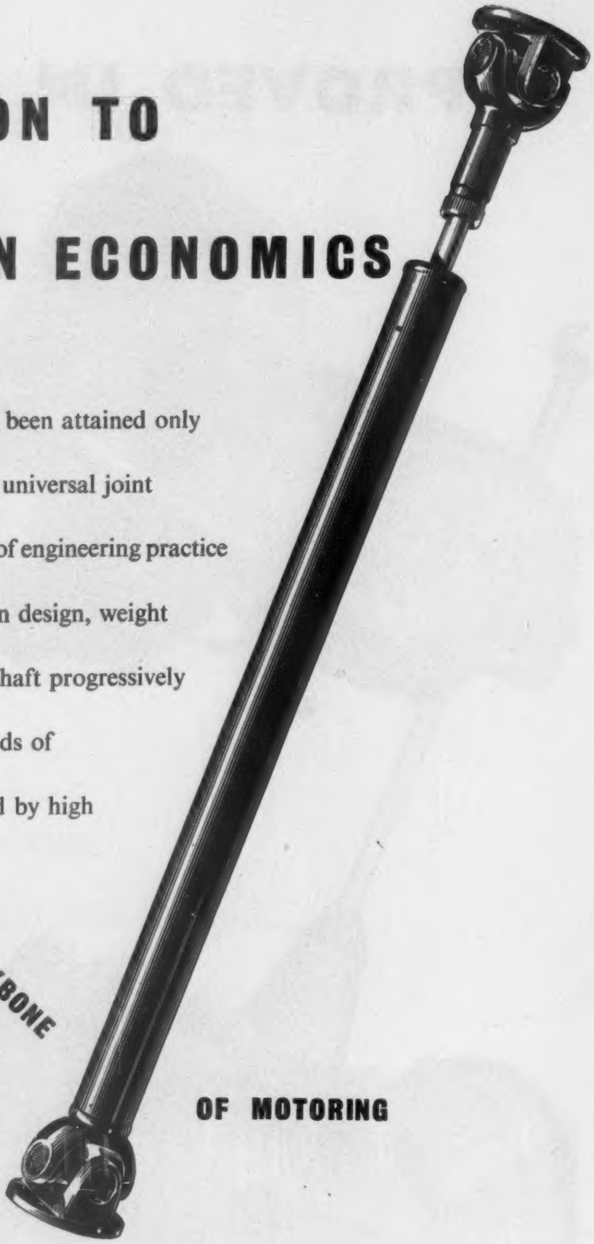
CORPORATION (SALES) LIMITED

AUTOMOBILE GEARBOX DIVISION

PARK WORKS HUDDERSFIELD

OUR CONTRIBUTION TO

MASS PRODUCTION ECONOMICS



Our objective of a standardised propeller shaft has been attained only because—ever since we introduced our mechanical universal joint thirty-one years ago—we have kept in the forefront of engineering practice both as to design and manufacturing techniques. In design, weight has been progressively reduced ; balancing of the shaft progressively improved ; stricter tolerances imposed ; the demands of increased engine efficiency always anticipated. And by high specialisation in manufacture we provide this precision component at a bedrock price.

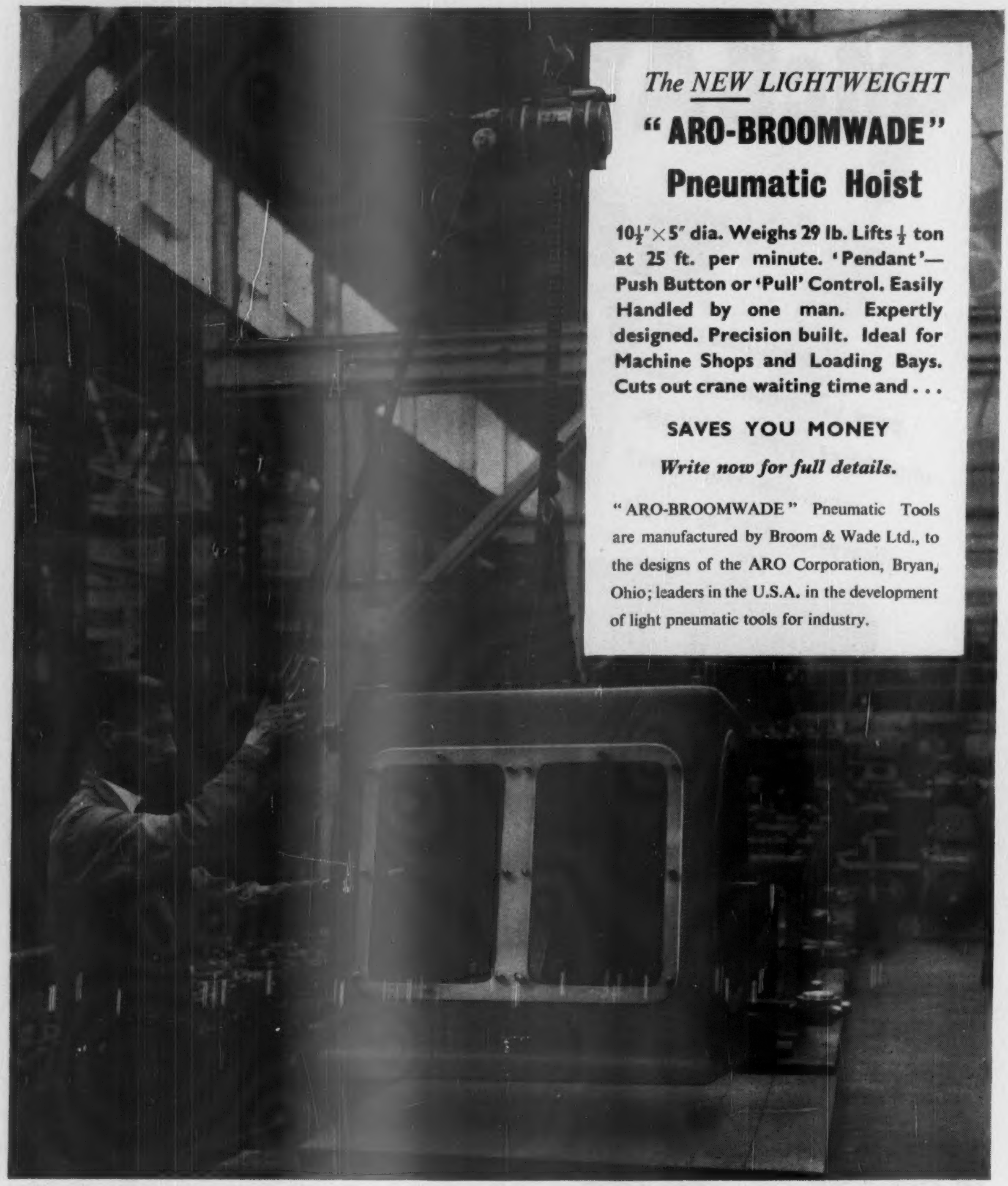
THE BACKBONE OF MOTORING

HARDY SPICER

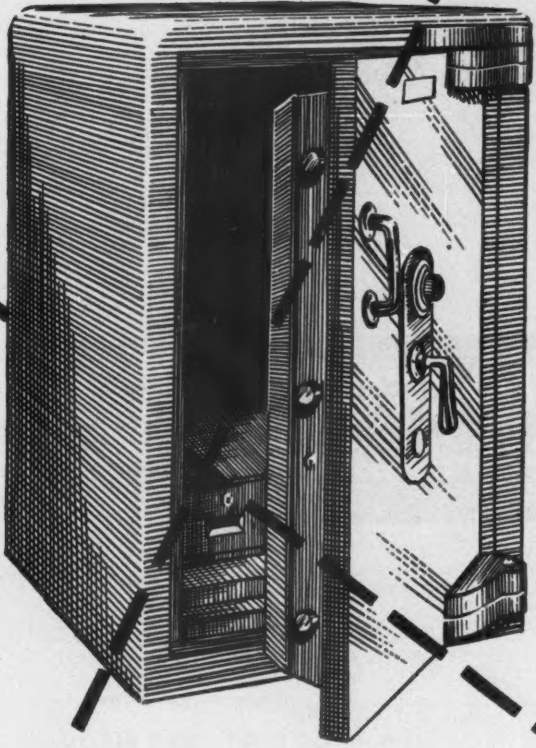
PROPELLER SHAFTS

PRECISION-BUILT FOR EFFICIENCY ★ MASS-PRODUCED FOR ECONOMY

HARDY SPICER LIMITED (A Birfield Company) BIRCH ROAD · WITTON · BIRMINGHAM 6


The NEW LIGHTWEIGHT
"ARO-BROOMWADE"
Pneumatic Hoist
10½″ × 5″ dia. Weighs 29 lb. Lifts ¼ ton at 25 ft. per minute. 'Pendant'—Push Button or 'Pull' Control. Easily Handled by one man. Expertly designed. Precision built. Ideal for Machine Shops and Loading Bays. Cuts out crane waiting time and . . .
SAVES YOU MONEY
Write now for full details.
"ARO-BROOMWADE" Pneumatic Tools are manufactured by Broom & Wade Ltd., to the designs of the ARO Corporation, Bryan, Ohio; leaders in the U.S.A. in the development of light pneumatic tools for industry.

We take no chances



We at Kirkstall have a reputation — a reputation for the care with which we choose the raw materials for our products; for the craftsmanship with which our Bright Steel Bars are produced; for the exhaustive way in which we test our bars. With such a reputation, we at Kirkstall take *no* chances!

There has been high quality iron or steel worked at Kirkstall for over seven hundred years.

Kirkstall Forge

Leeds • Yorkshire

TELEPHONE: HORSFORTH 2821

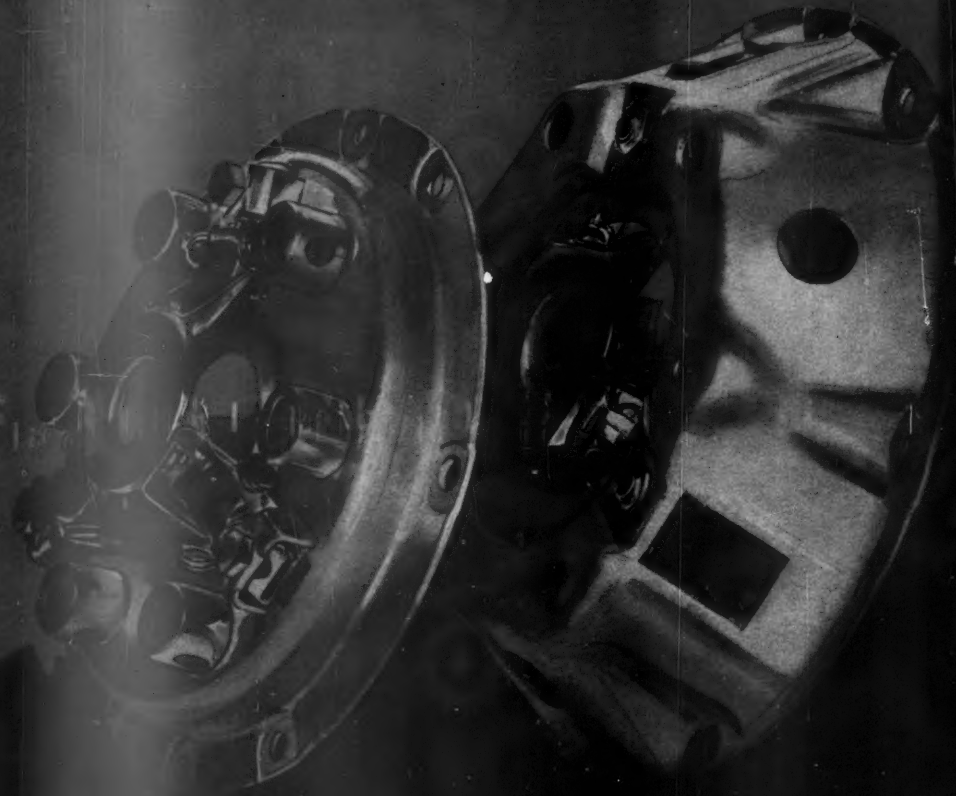
M 27

In

Plain carbon or low all
frequently replace mo
highly alloyed steels wh
hardening is used.

If you have a
*problem—**EFCO***
have the answer

EFCO



Some of the Borg & Beck range of friction clutches, covering every type of car.

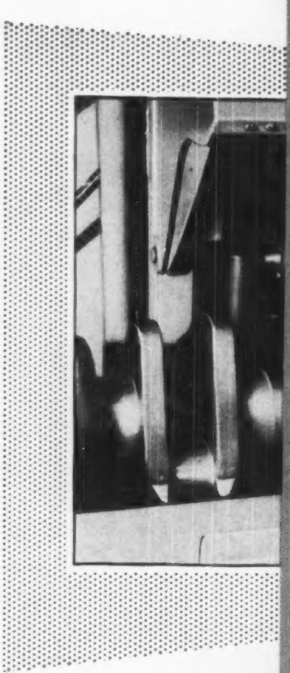
Below: the spring-centre driven plate, which gives perfect transmission of power.

BORG & BECK Registered Trade Mark

BORG & BECK CO LTD

LOCKHEED HYDRAULIC BRAKE CO LTD

AUTOMOTIVE PRODUCTS CO LTD



In

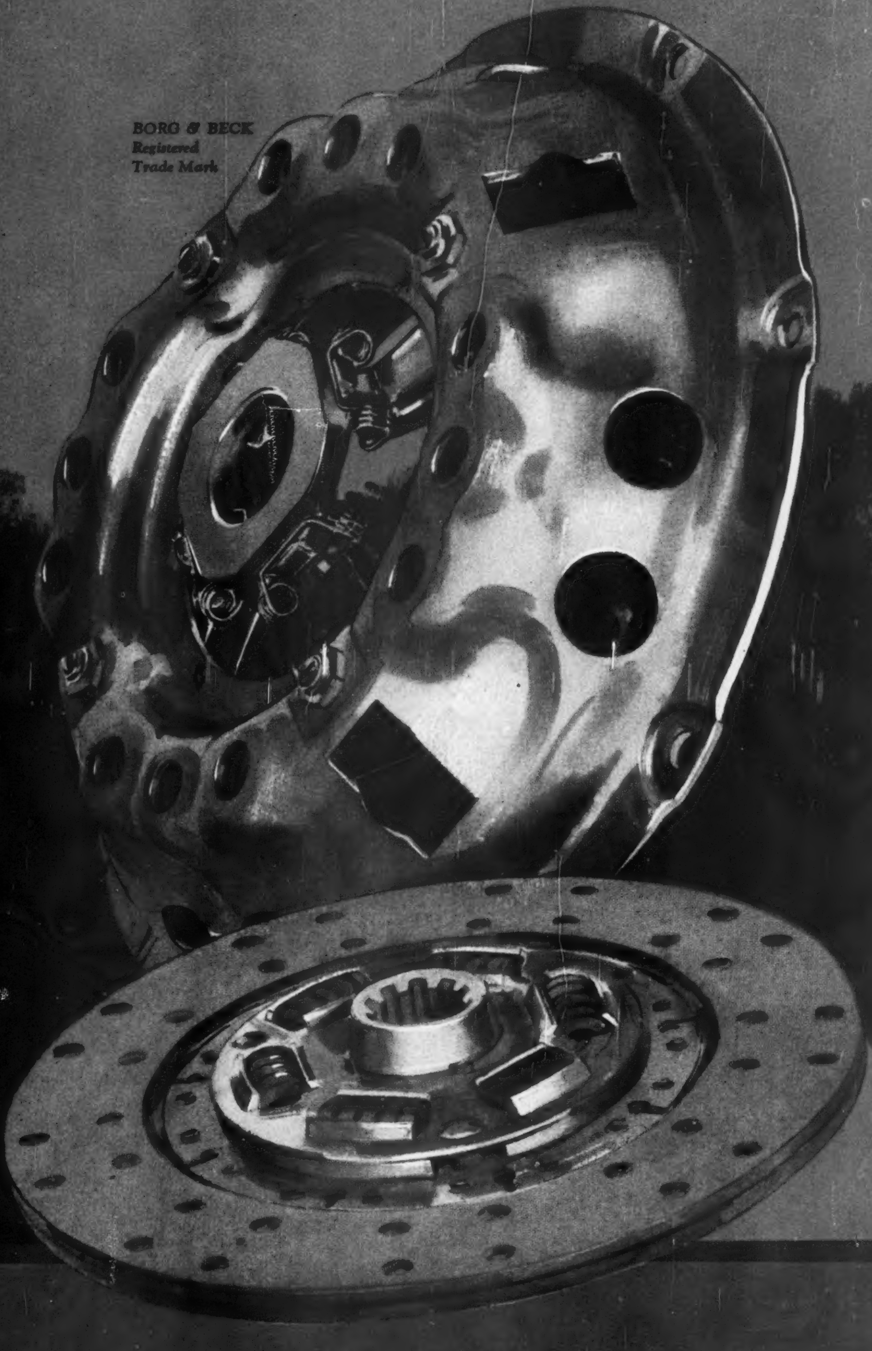
Plain carbon or low all
frequently replace mor
highly alloyed steels wh
hardening is used.

If you have a h
*problem—**EFCO***
have the answer

EFCO

Some of the Borg & Beck range of friction clutches, covering every type of car.

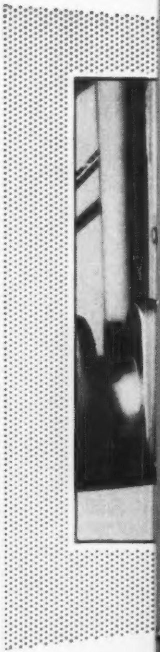
Below: the spring-centre driven plate, which gives perfect transmission of power.

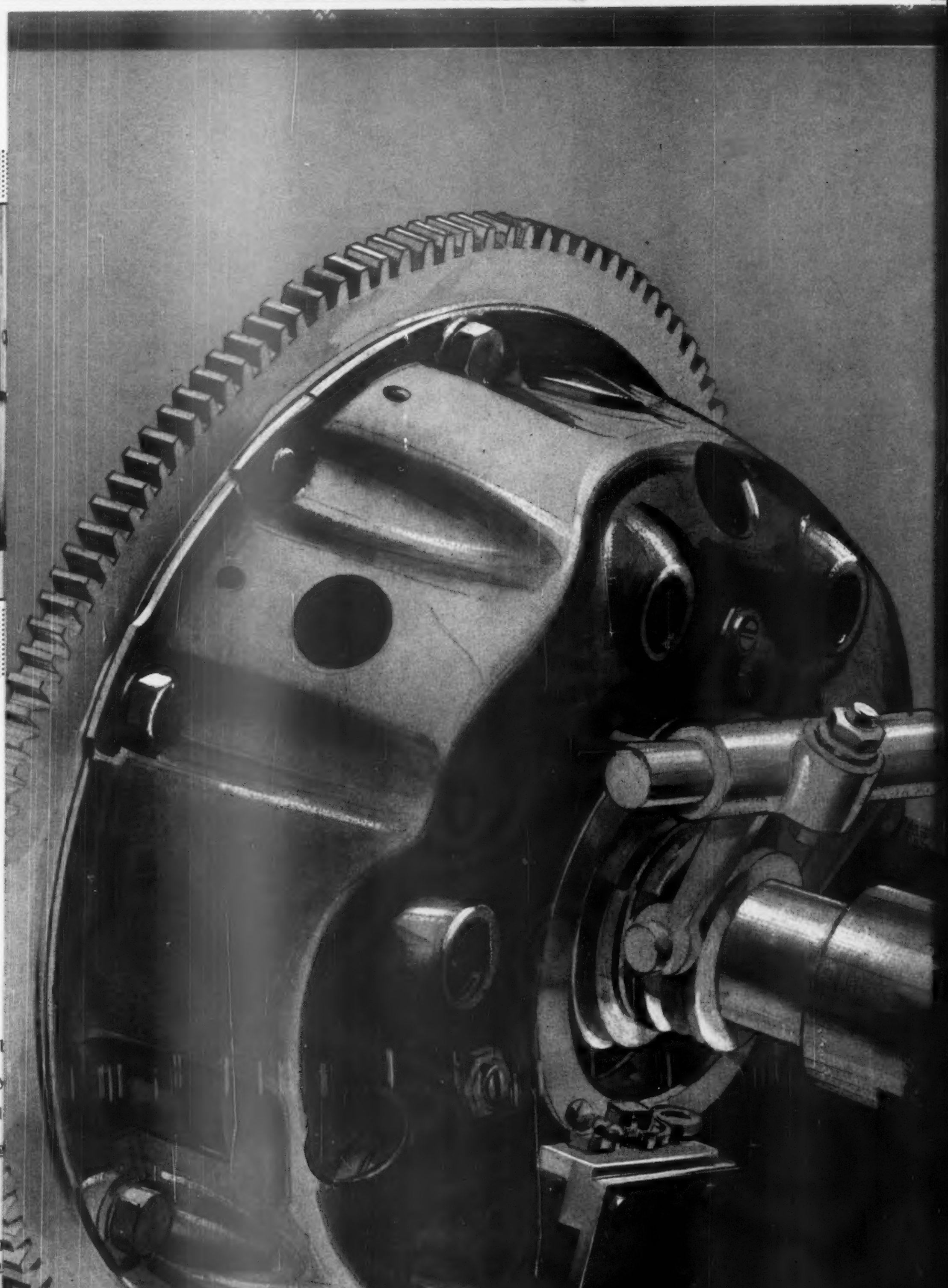


BORG & BECK CO LTD

LOCKHEED HYDRAULIC BRAKE CO LTD

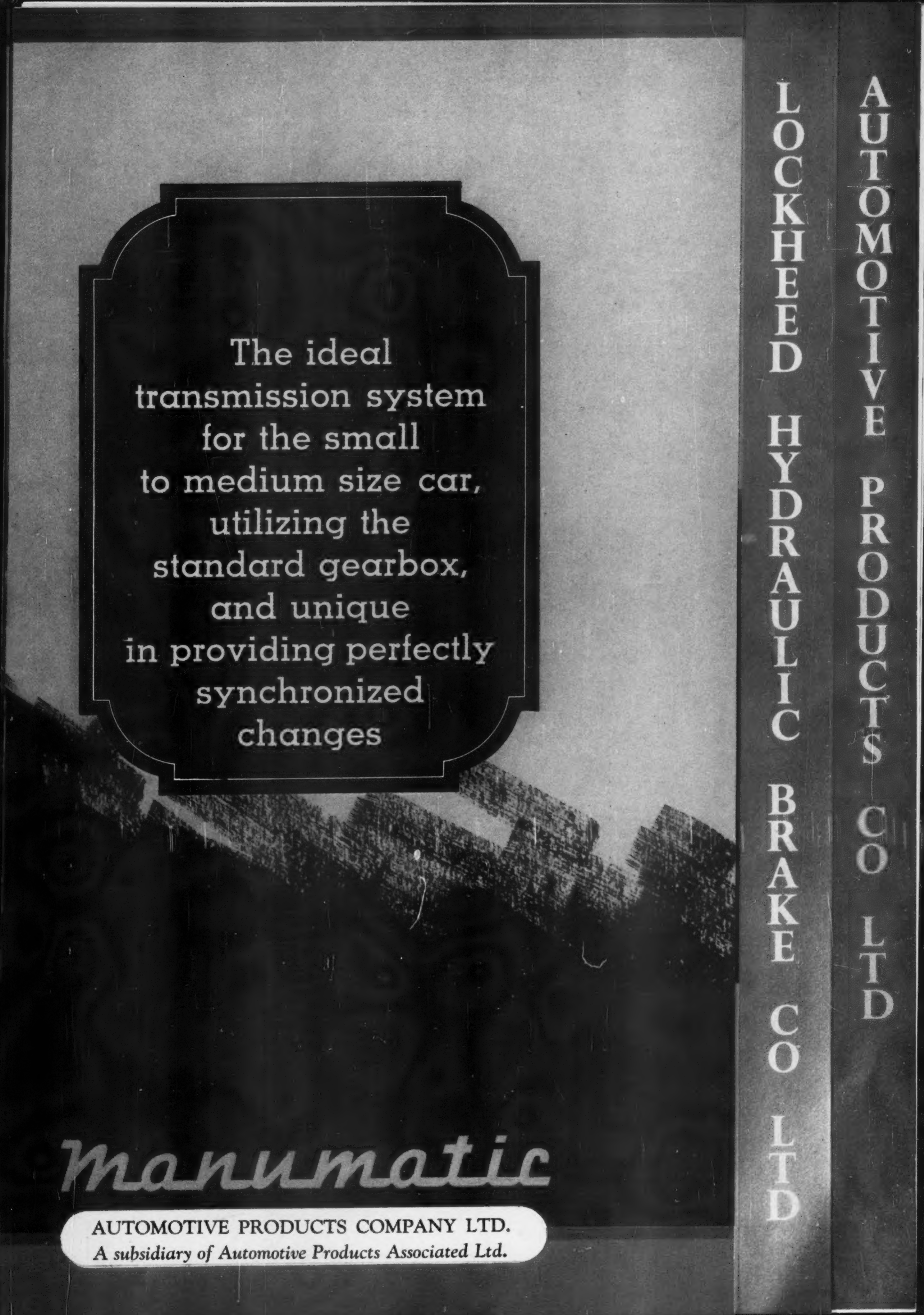
AUTOMOTIVE PRODUCTS CO LTD

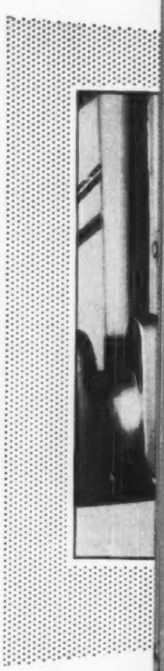




Plain carb
frequently
highly all
hardening

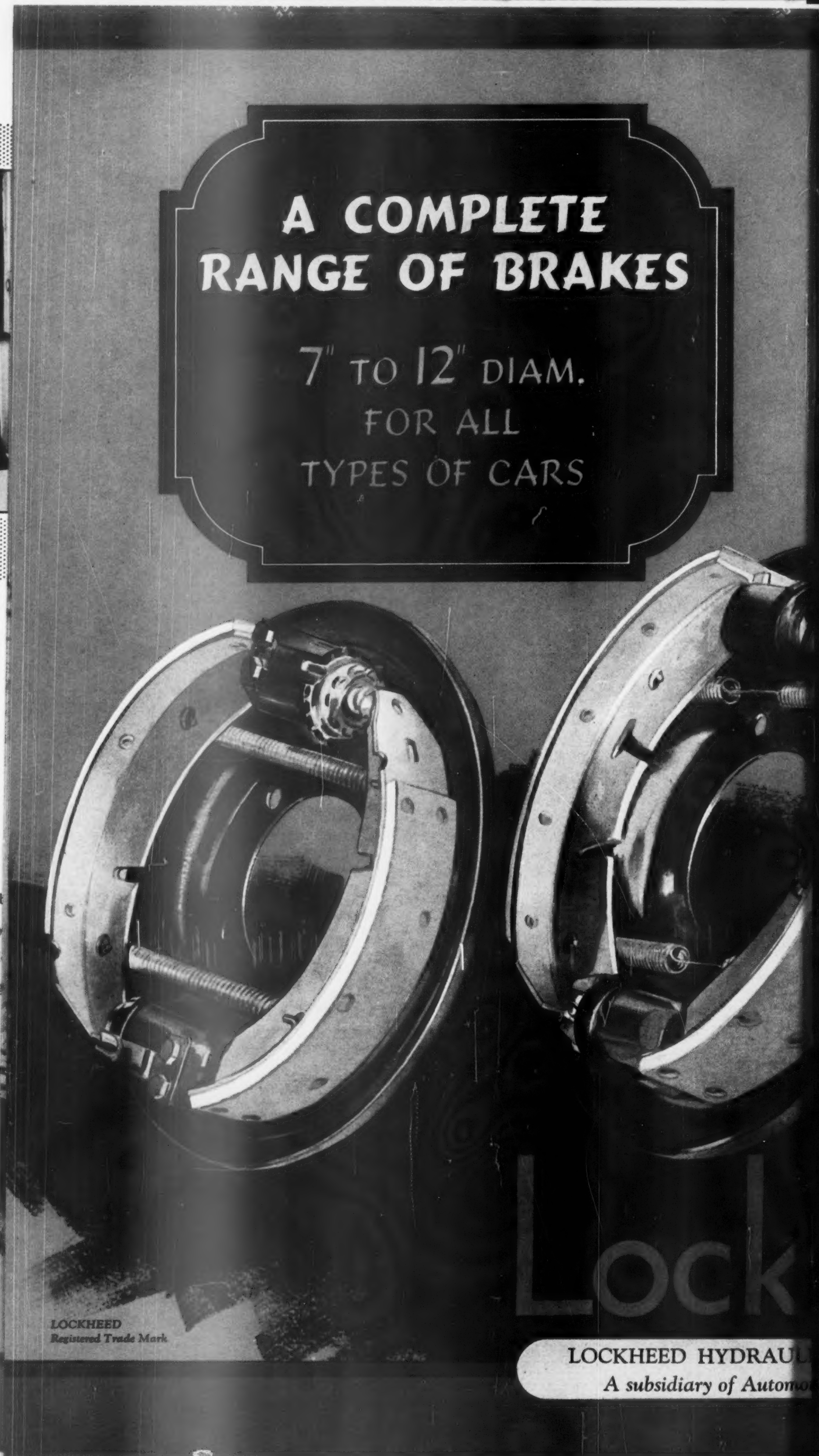
If you
problem
have th

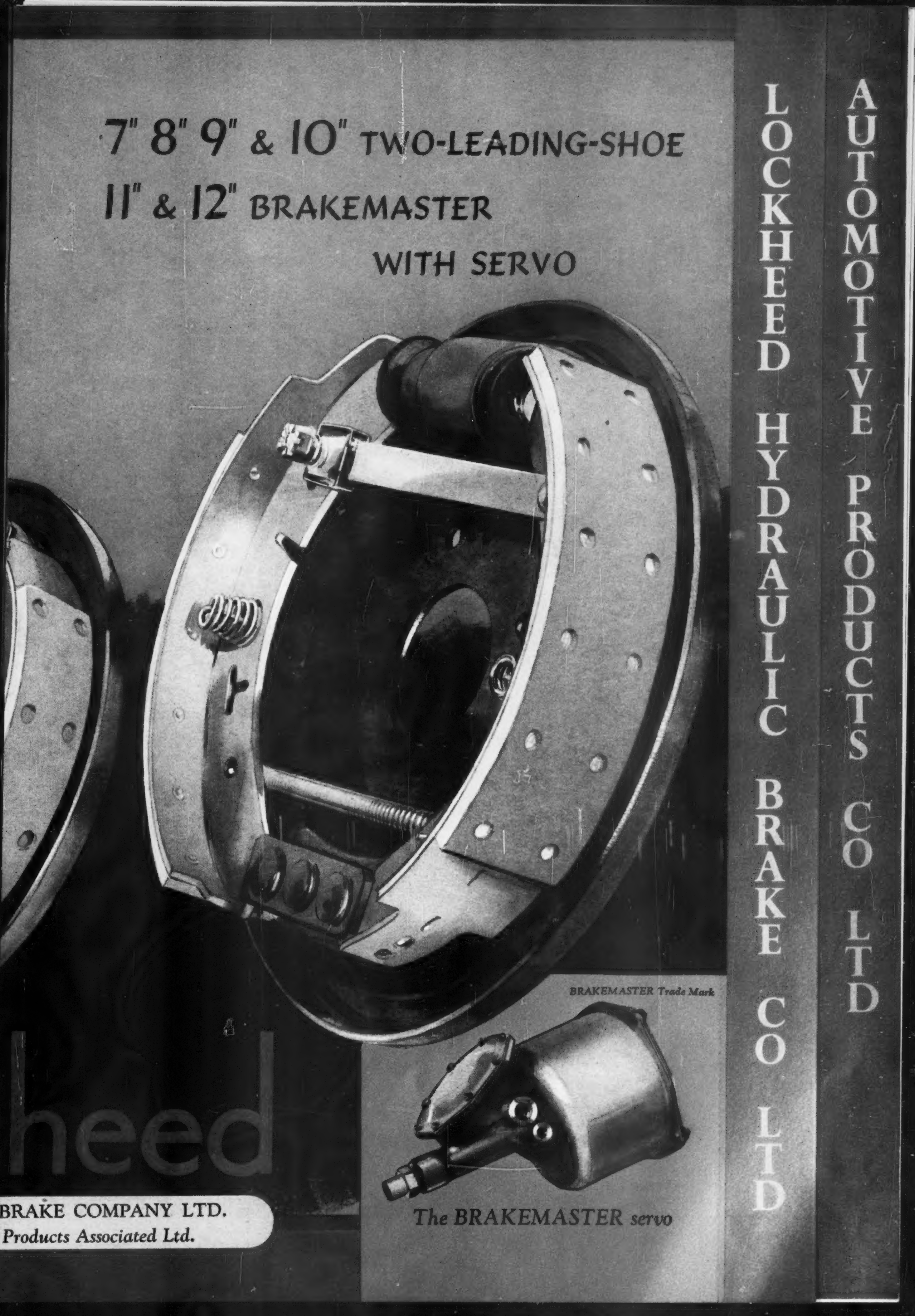

The ideal transmission system for the small to medium size car, utilizing the standard gearbox, and unique in providing perfectly synchronized changes
Manumatic
AUTOMOTIVE PRODUCTS COMPANY LTD.
A subsidiary of Automotive Products Associated Ltd.
LOCKHEED HYDRAULIC BRAKE CO LTD
AUTOMOTIVE PRODUCTS CO LTD



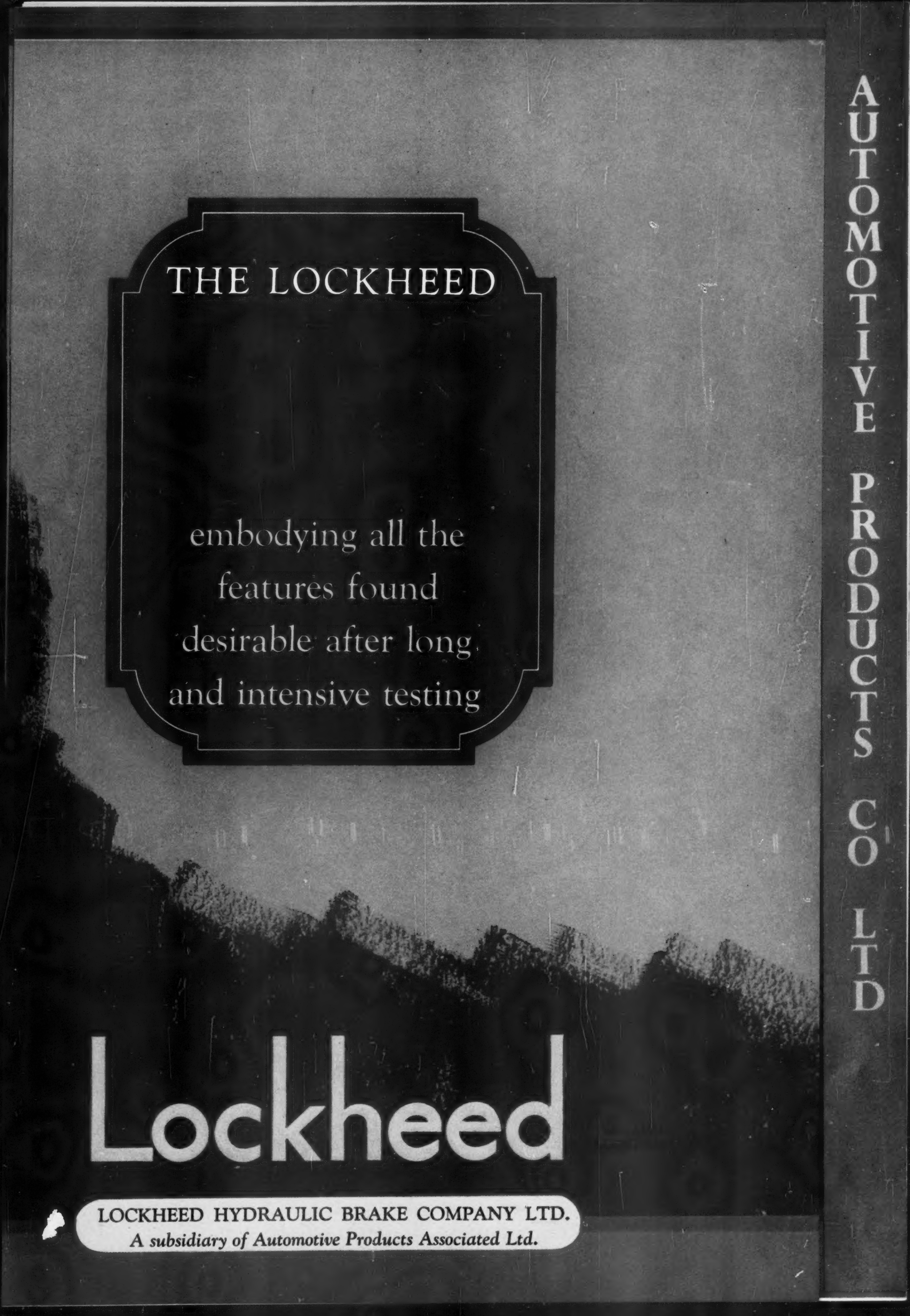
Plain carb
frequently
highly all
hardening

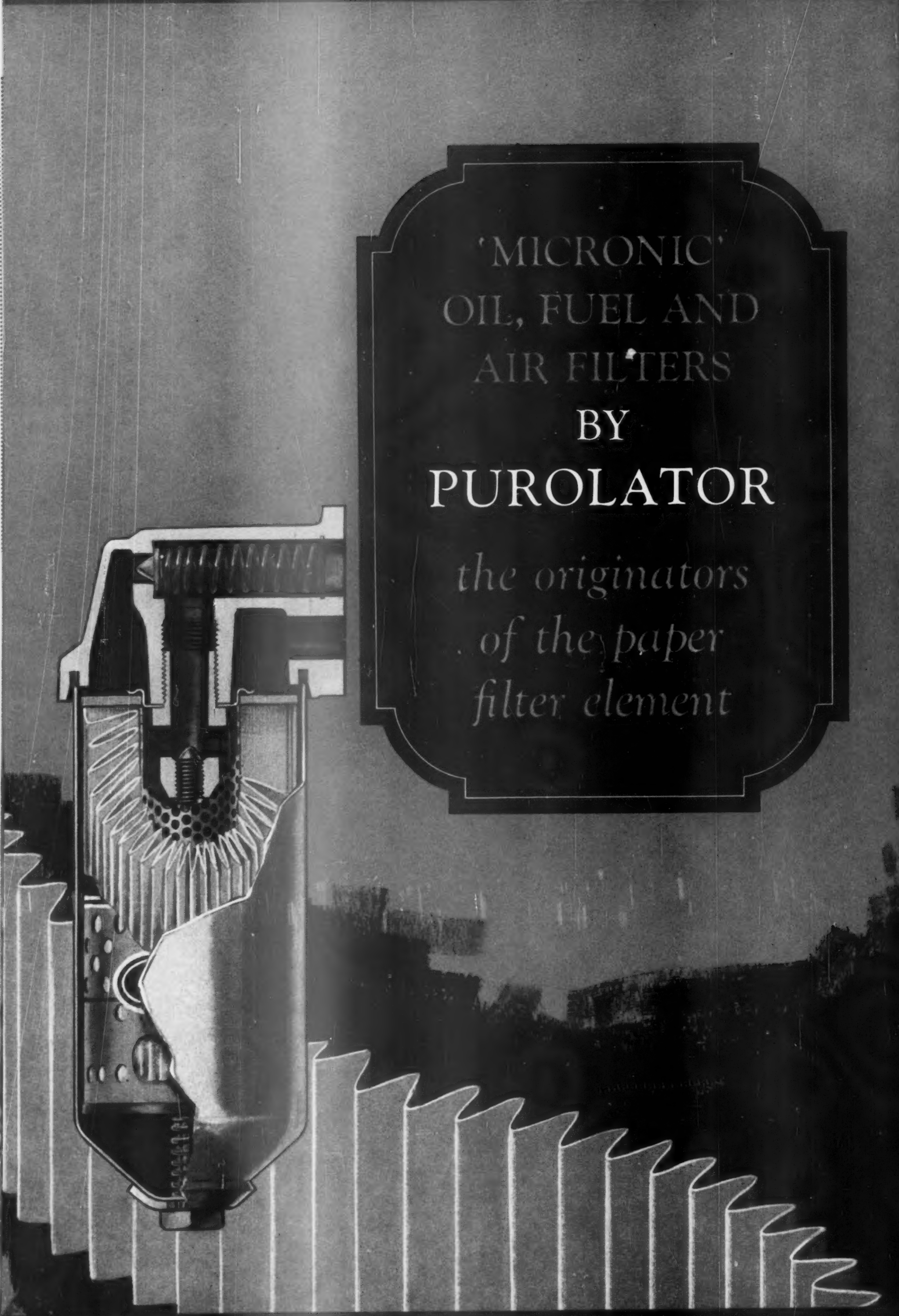
If you
problem
have th





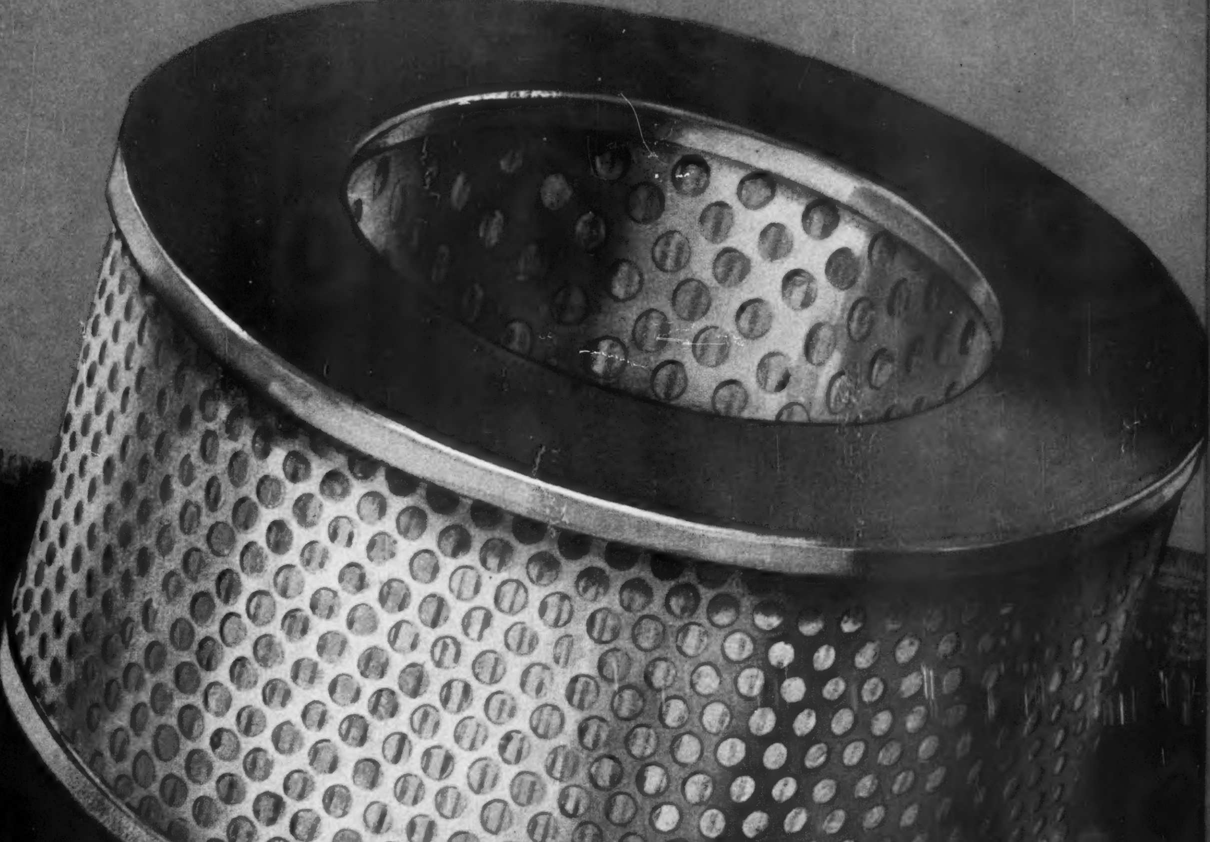
7" 8" 9" & 10" TWO-LEADING-SHOE
11" & 12" BRAKEMASTER
WITH SERVO
heed
BRAKE COMPANY LTD.
Products Associated Ltd.
BRAKEMASTER Trade Mark
The BRAKEMASTER servo
LOCKHEED HYDRAULIC BRAKE CO LTD
AUTOMOTIVE PRODUCTS CO LTD


THE LOCKHEED
embodying all the features found desirable after long and intensive testing
Lockheed
LOCKHEED HYDRAULIC BRAKE COMPANY LTD.
A subsidiary of Automotive Products Associated Ltd.
AUTOMOTIVE PRODUCTS CO LTD


'MICRONIC'
OIL, FUEL AND
AIR FILTERS
BY
PURLATOR
the originators
of the paper
filter element

Below: the filter element of the new Purolator 'Micronic' dry-type air filter—backed by many million miles of road service

Opposite: a typical full-flow oil filter

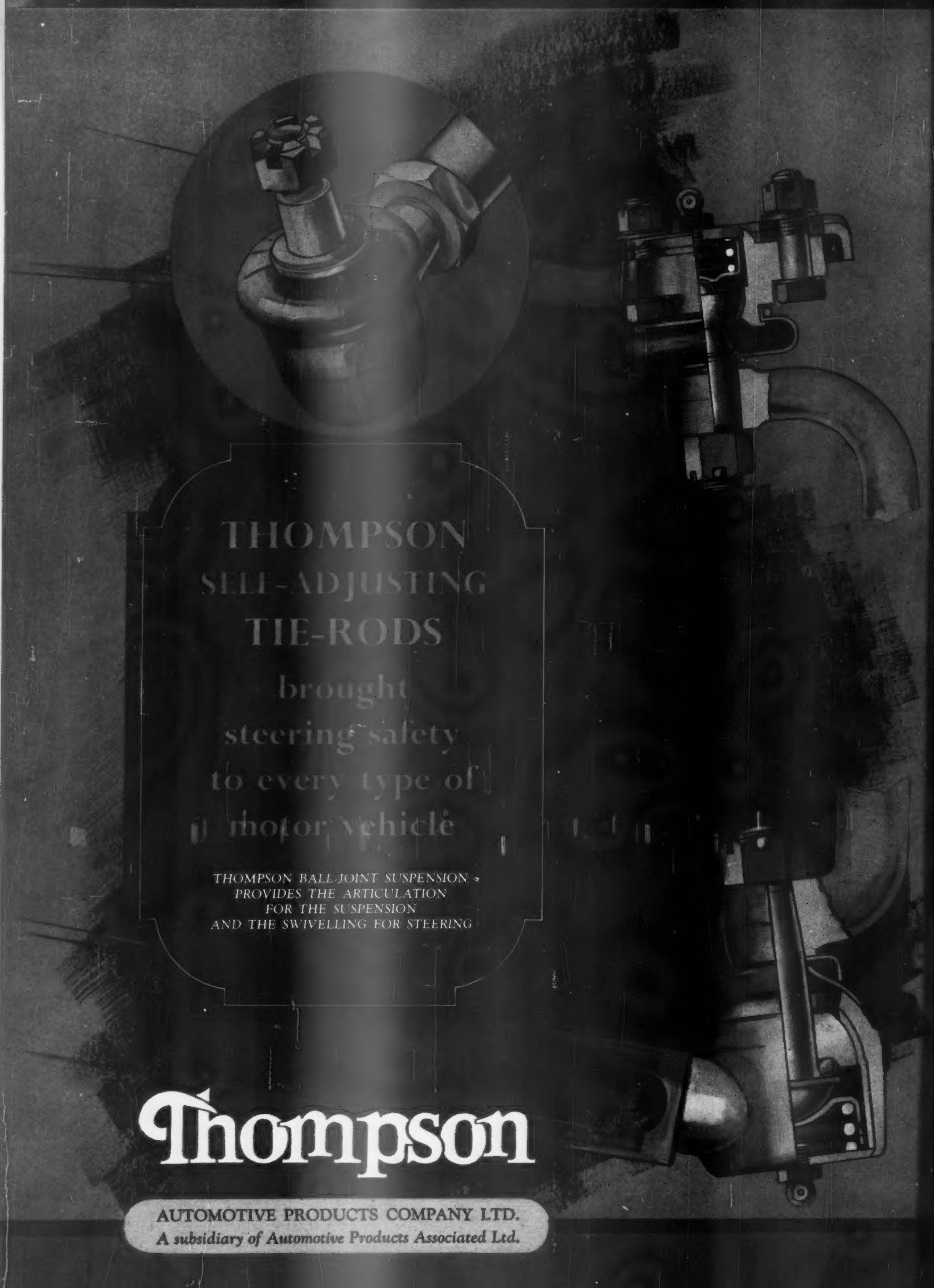


PUROLATOR

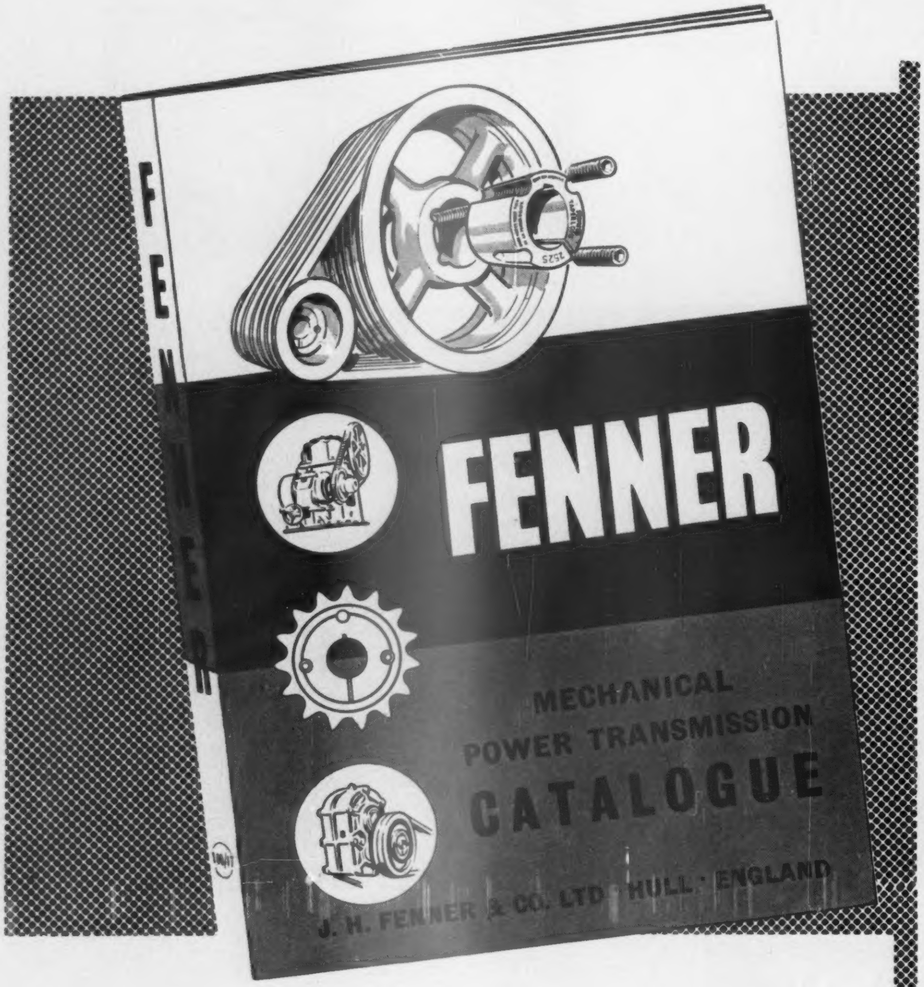
AUTOMOTIVE PRODUCTS COMPANY LTD.

A subsidiary of Automotive Products Associated Ltd.

AUTOMOTIVE PRODUCTS CO LTD


THOMPSON
SELF-ADJUSTING
TIE-RODS
brought
steering safety
to every type of
motor vehicle
THOMPSON BALL-JOINT SUSPENSION
PROVIDES THE ARTICULATION
FOR THE SUSPENSION
AND THE SWIVELLING FOR STEERING
Thompson
AUTOMOTIVE PRODUCTS COMPANY LTD.
A subsidiary of Automotive Products Associated Ltd.

Designing a Drive?



V-BELT DRIVES

V-BELTS

TAPER-LOCK PULLEYS

TAPER-LOCK CHAIN DRIVES

TORQUE ARM SPEED REDUCERS

VARIABLE SPEED DRIVES

COUPLINGS

CLUTCHES

ANTI-VIBRATION MOUNTINGS

FLAT BELTINGS

PACKINGS AND SEALS

•

Britain's first pocket size drive handbook, with copious information, convenient to handle and easy to read. Sales 'patter' has been rigorously excluded—this is a working tool for designers and draughtsmen who have already decided to buy Fenner products. It contains just the information the user requires as determined in 25 years experience by the engineering staff of

Set in this modern Rockwell typeface and printed by the lithographic process, this catalogue is a model of clarity even in its $7\frac{1}{2}" \times 5\frac{1}{2}"$ size.

J · H · FENNER & CO · LTD · HULL

LARGEST MAKERS OF V-BELT DRIVES IN THE COMMONWEALTH

Three frozen cheers for Electrolux

H'rm . . . H'rm ! — said the MD clearing his throat : I have great pleasure in proposing a toast *(CHEERS)* to the Electrolux boys *(LOUD CHEERS)* on the production at Luton of their millionth fridge *(IMMENSE AND PROLONGED APPLAUSE).*

Ladies and gentlemen, only a company of immense skill and enthusiasm in every department could make a million refrigerators for a people who are undoubtedly more cold more often and at more unlikely seasons than any nation in the world. Such a company is the Electrolux Company ! *(MORE CHEERS mixed with cries of BRING ON THE TOAST).* If proof of the far sightedness, the deep wisdom, the high technical intelligence of this Company be asked, I need only say that in every department of Electrolux only the best men, the best materials and THE BEST TOOLS ARE GOOD ENOUGH — AS IS PROVED *(he cried, raising his voice to a fortissimo)* BY THEIR INCREASING USE OF

DESOUTTER TOOLS!

(loud cries of "Oh . . . you self-advertising cad !" and "Push him in the fridge !")

DESOUTTER BROS. LIMITED, THE HYDE, HENDON, N.W.9. TEL: COLINDALE (5 LINES) GRAMS: DESPNUCO HYDE, LONDON.
CRC286

In the Gisholt SUPERFINISHING machine, as the work rotates it is subjected to a light-pressure contact of an oscillating abrasive stone. This imparts a scrubbing effect which removes minor surface irregularities such as chatter and feed marks and "smear" metal surface left by the usual grinding operation. Surface finishes of one micro-inch can be quickly and economically attained by Superfinishing.

SUPERFINISHING MACHINES

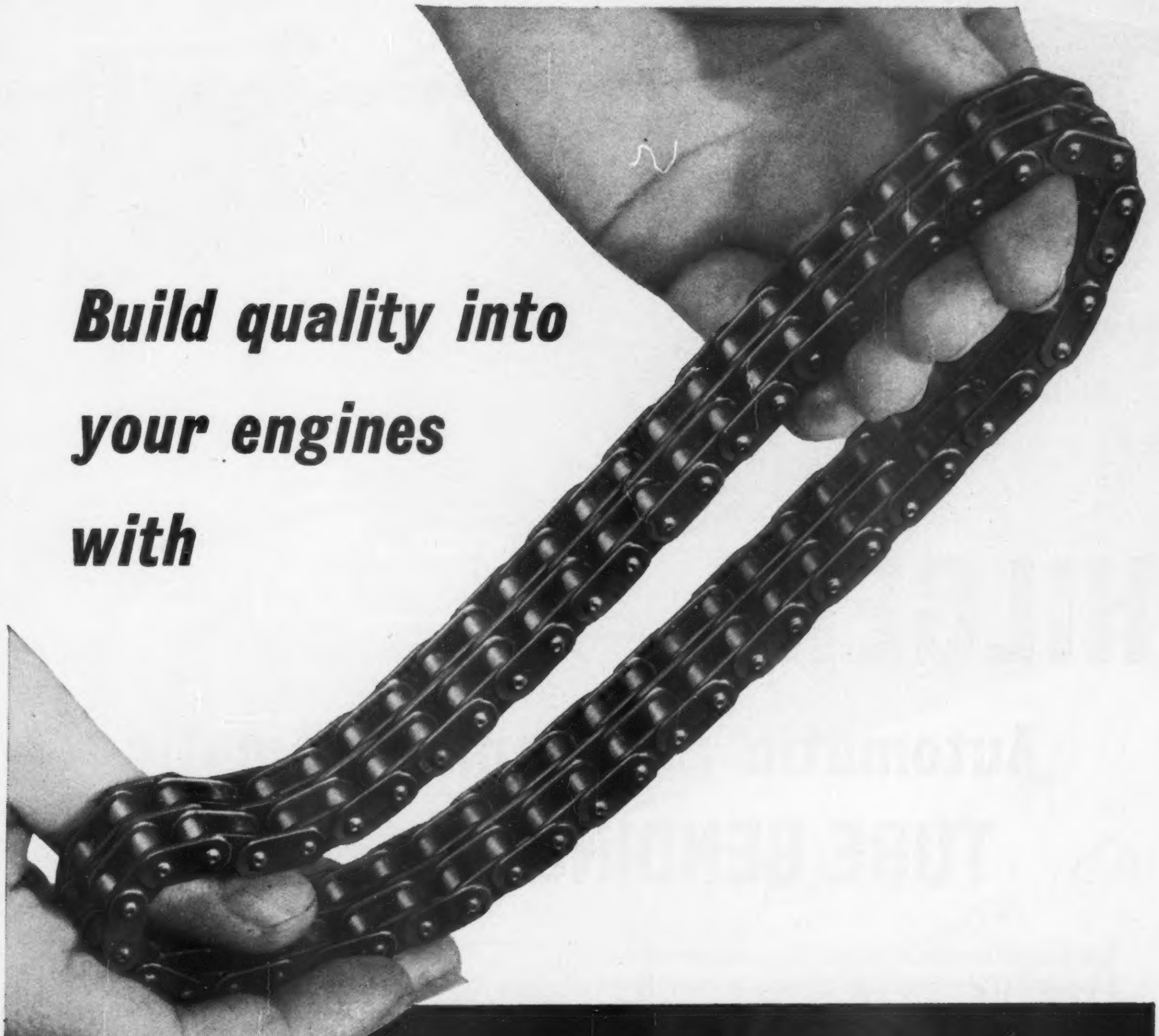
Gisholt General Purpose Superfinisher Model 52A is shown for work up to 10" swing, 36" between centres, maximum weight 125 lb.

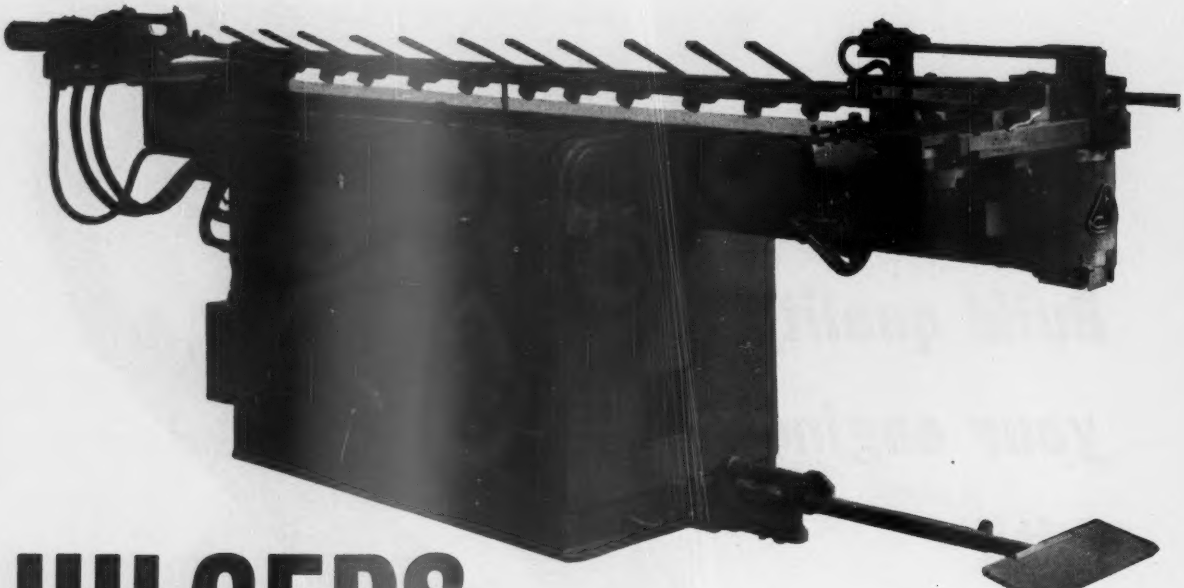
The No. 3 Superfinishing Attachment is designed for tool post mounting on engine or turret lathes; for work up to 9" diameter.

BALANCING MACHINES

SPECIFICATIONS	HORIZONTAL—FLOOR TYPE			
	1S	13S	3S	31S
Work Capacity, Weight in lbs.	1–30	2–50	15–300	2-300
Overall Diameter	12"	24"	24"	24"
Shaft Diameter at Bearing Surfaces	$1\frac{1}{8}$"	$2\frac{1}{8}$"	5"	$2\frac{1}{8}$" & 5"
Maximum Distance Between Bearings	12"	24"	24"	24"
Balancing Speeds, R.P.M.	1000–3000	1000–2000	1000–2000	1000–2000
Floor Space (Approx.)	43" × 43"	68" × 43"	68" × 43"	68" × 43"
Net Weight, lbs. (Approx.)	1500	1700	1800	1900

Gisholt DYNETRIC Type S Balancing machines provide a means for quickly and accurately measuring and locating unbalance in parts weighing from a few ounces to several hundred pounds. They are equally suitable for either large or small quantities of similar parts. The required amount of correction to balance is indicated in practical units such as in thousands of an inch depth of drill, in $\frac{1}{64}$ inch lengths of wire solder, or in any other units most satisfactory for the specific workpiece.


Build quality into
your engines
with
RENOLD
TIMING CHAINS
RENOLD —the FIRST name in precision chain
RENOLD CHAINS LIMITED MANCHESTER



HILGERS

Automatic and semi-automatic TUBE BENDING MACHINES

The complete range of Hilgers Semi-Automatic and Automatic Tube Bending Machines is suitable for both right and left hand bending, and working speed is infinitely variable. When the machine is adjusted to lowest working speed the return movement always switches automatically to the highest return speed. Moving parts are completely enclosed and lubricated automatically by a central system, thereby cutting maintenance to the minimum. Many of these excellent machines can be seen in operation in the United Kingdom.

BRIEF SPECIFICATIONS

Type HE.38 Semi-Automatic Cold Tube Bending Machine with Combined Mechanic Hydraulic Drive.

Largest tube diameter approx. 1½"o.d. x 14 s.w.g. thick
Number of bends per hour......................up to 250

Types HYB.50 & HYB.70 Hydraulic Automatic Cold Tube Bending Machines

	HYB.50	HYB.70
Largest tube diameter	approx. 2" o.d. x 5/64" thick	approx. 2¾" o.d. x 5/64" thick
Number of bends per hour	up to 600	up to 450

Type HDY.114 Semi-Automatic Cold Tube Bending Machine with Hydraulic Operation.

Largest tube diameter
114 mm. o.d. x 5 mm. wall thickness
Number of bends per hour.....................up to 400

Selson
MACHINE TOOLS

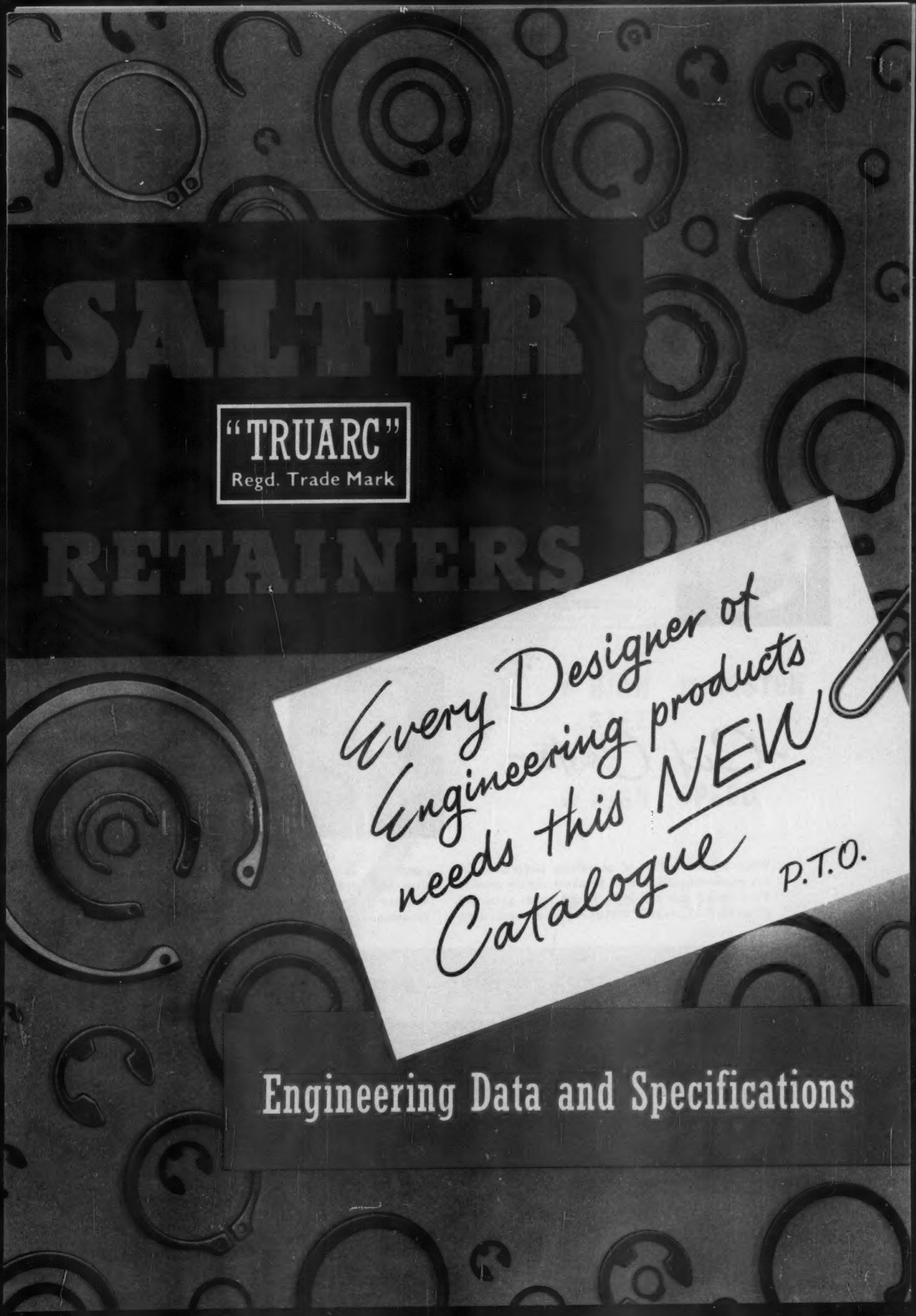
SOLE AGENTS

The Selson Machine Tool Co. Ltd

41-45 MINERVA ROAD, NORTH ACTON, LONDON, N.W.10

THE 600 GROUP

Telephone: Elgar 4000 (10 lines)   Telegrams: Selsomachi, London, N.W.10

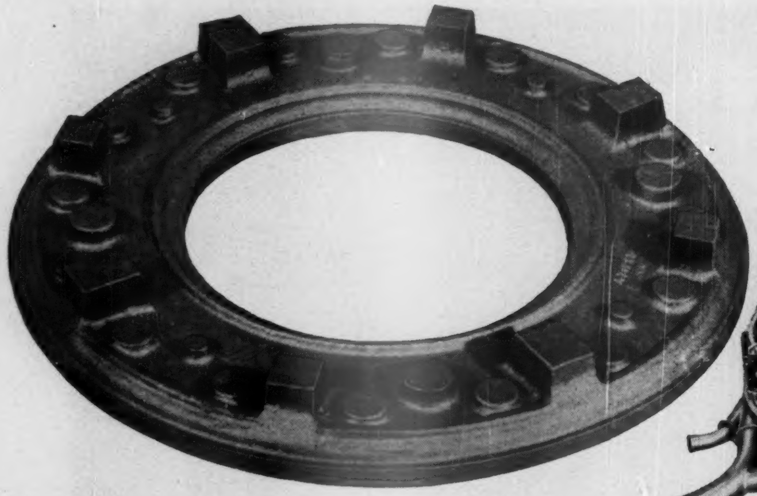

SALTER
"TRUARC"
Regd. Trade Mark
RETAINERS
Every Designer of Engineering products needs this NEW Catalogue
P.T.O.
Engineering Data and Specifications

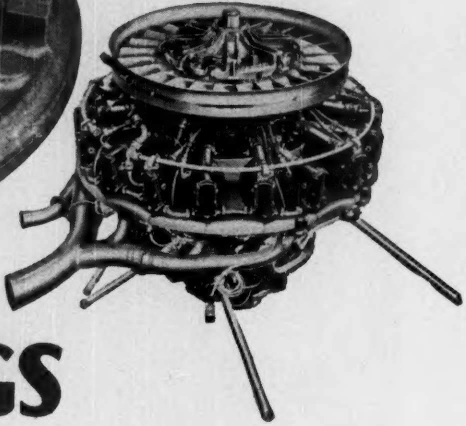

Buy
SPEEDICUT
Britain's
Best
Hacksaw
SPEEDICUT HIGH SPEED STEEL
SPEEDICUT
● HIGH TUNGSTEN STEEL
● HIGH SPEED
● LONG LIFE
SPEEDICUT
The Hallmark of Engineers' Cutting TOOLS
Made in Britain's Largest Tool Factory; FIRTH BROWN TOOLS LTD. SHEFFIELD

Dunlop Technicians will always advise on any problems concerning installation of Flexible Pipes, write please to :—

DUNLOP RUBBER COMPANY LIMITED · ST. GEORGE'S ROAD · COVENTRY · TELEPHONE: COVENTRY 64171

Alvis Leonides Engines



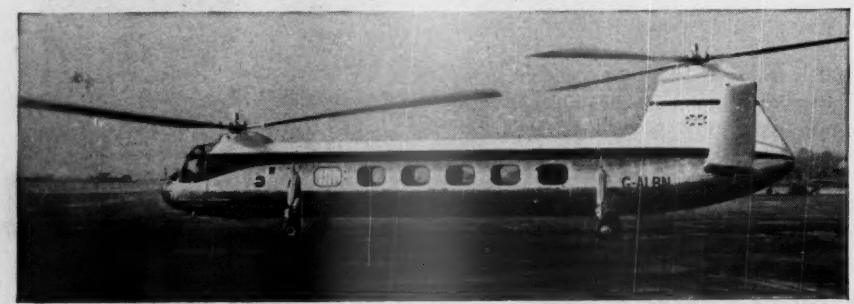


use

HARPER CASTINGS

Harpers make the clutch plate castings used in Alvis Leonides engines, widely installed in helicopters. High strength, soundness and freedom from distortion are essential, as these castings provide the means by which the power is transmitted from the engine to the rotor blades.
Every casting is radiologically examined by Harpers to meet the requirements of the D.A.I.

Harper quality covers Grey Iron, Ductile Iron and Meehanite castings, also metal pressings and other finishes and sub-assembly work.

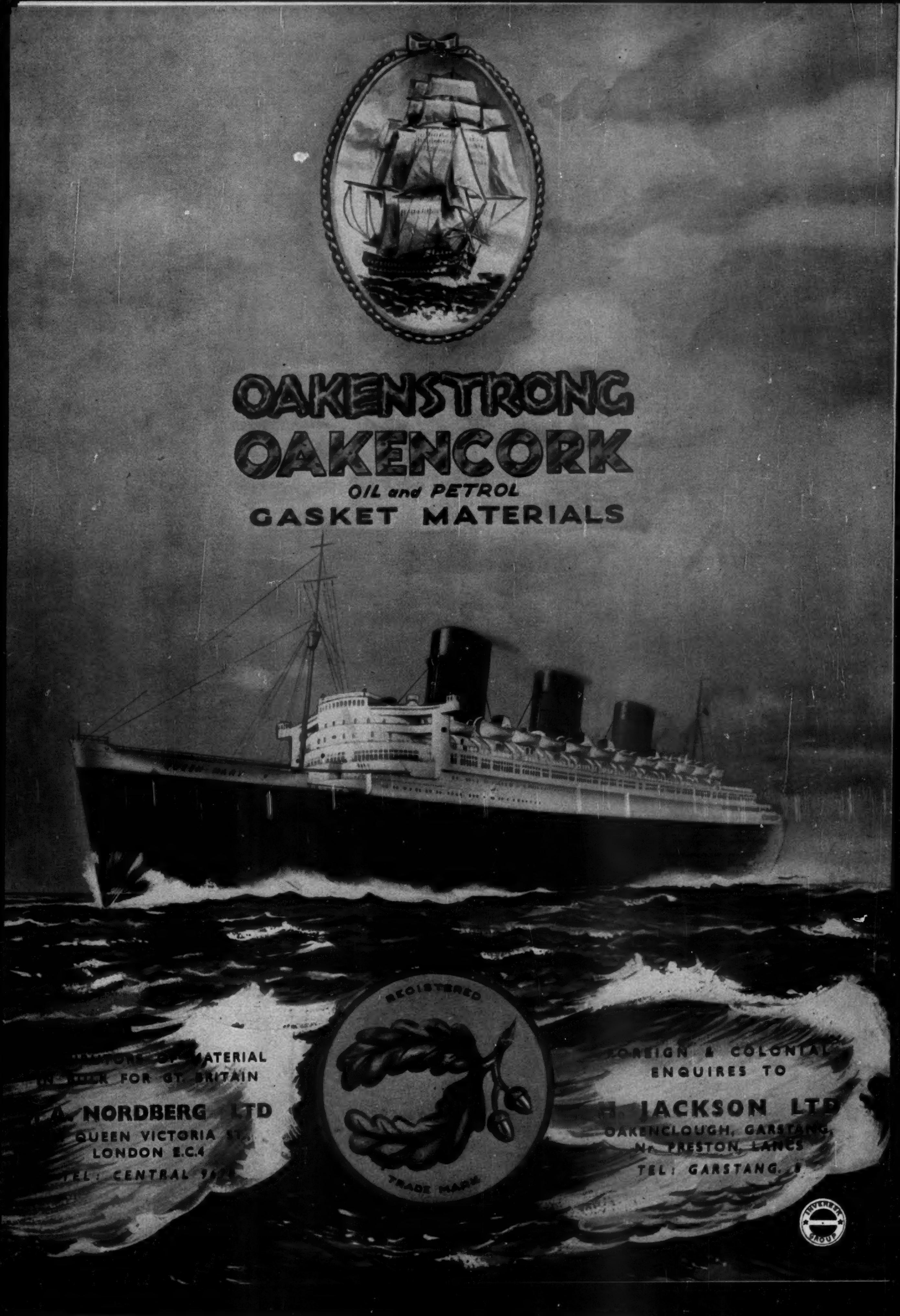


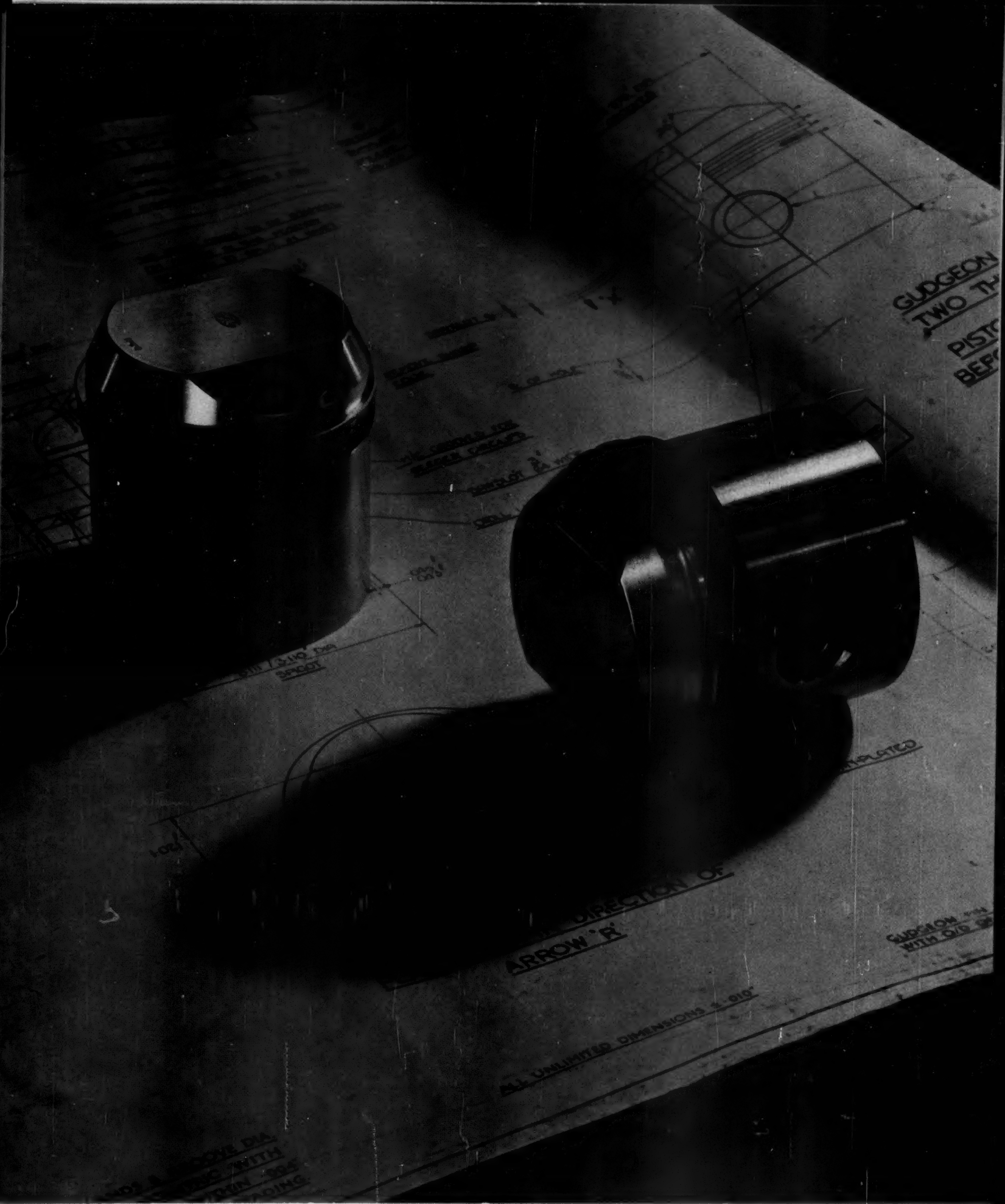
JOHN HARPER & CO. LTD. JOHN HARPER (MEEHANITE) LTD.
ALBION WORKS Phone: WILLENHALL 124 (5 lines) Grams: HARPERS, WILLENHALL **WILLENHALL**

CASTINGS WITH THE HARPER SKIN

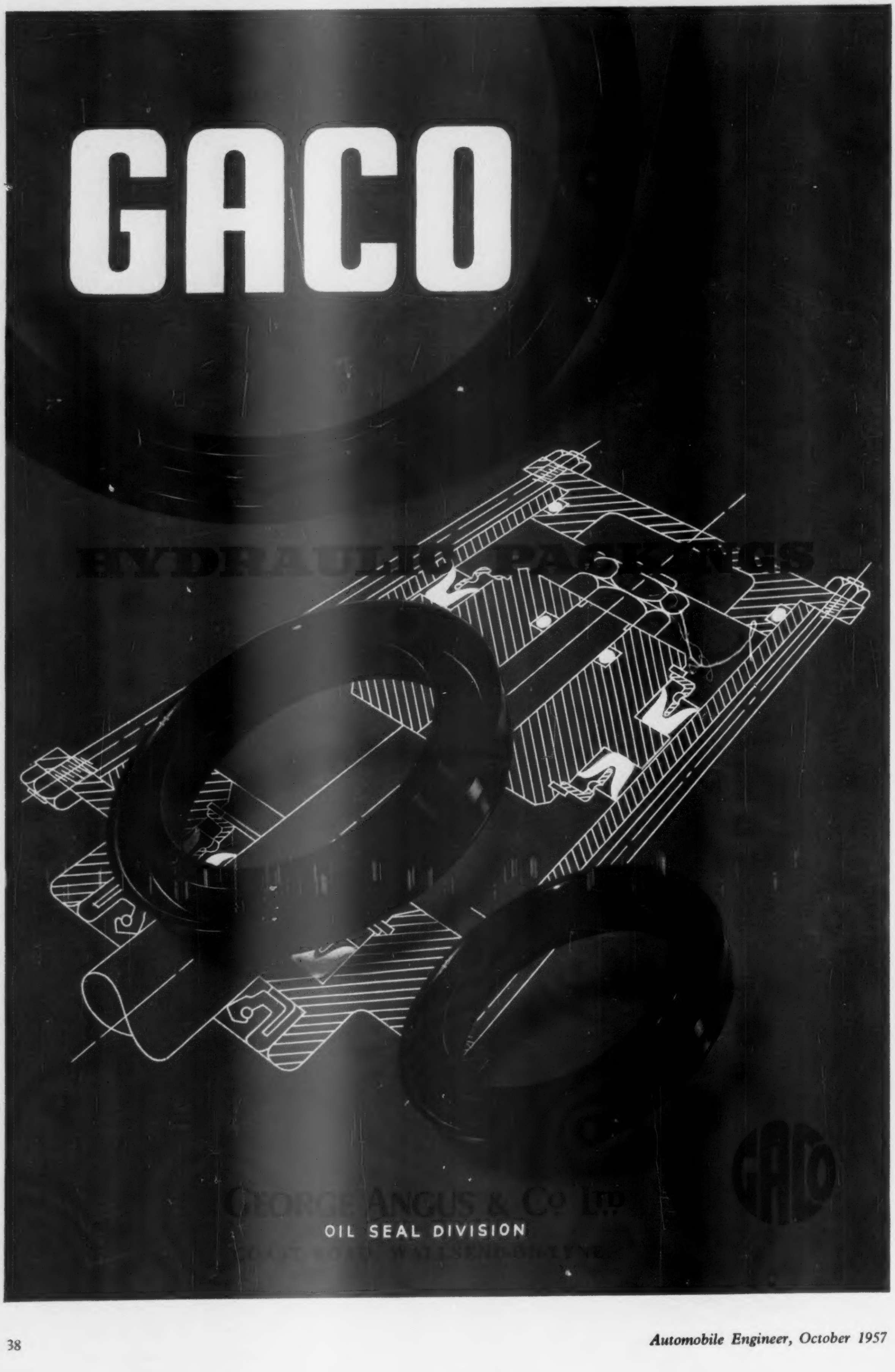
LONDON OFFICE: SEAFORTH PLACE, 57, BUCKINGHAM GATE, LONDON S.W.1 Tel.: TATE GALLERY 0286

H621 MANCHESTER OFFICE: c/o B. J. Brown & Partners Ltd. 248/9 Royal Exchange, Manchester 2


OAKENSTRONG
OAKENCORK
OIL and PETROL
GASKET MATERIALS
REGISTERED
TRADE MARK
FOR GT. BRITAIN
NORDBERG LTD
QUEEN VICTORIA ST.
LONDON E.C.4
FOREIGN & COLONIAL
ENQUIRES TO
JACKSON LTD
OAKENCLOUGH, GARSTANG,
Nr. PRESTON, LANCS
TEL: GARSTANG.
INVERESK GROUP


GUDGEON
DIRECTION OF
ARROW 'R'
ALL UNLIMITED DIMENSIONS ±·010"


GUDGEON
ARROW 'R'


GACO
HYDRAULIC PACKINGS
GEORGE ANGUS & Co LTD
OIL SEAL DIVISION

We like to think...

We like to think we know as much about heat-treatment as most people. After all, we've been in the business of designing and making furnaces for nearly forty years. Take forced air circulation for example. We pioneered the use of the centrifugal fan, and forced air circulation is still the best available medium for tempering, secondary hardening, and the heat-treatment of light alloys and non-ferrous annealing. Our furnaces permit precise control of temperature, with simplicity of operation and maintenance. We'll be glad to supply further information and make recommendations to meet your needs.

FORCED AIR CIRCULATION FURNACES

ELECTRIC WILD BARFIELD FURNACES

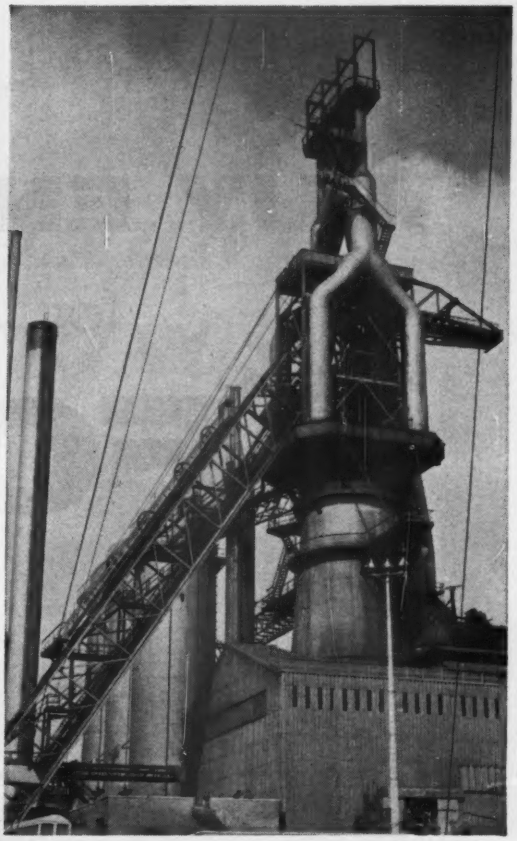
FOR ALL HEAT-TREATMENT PURPOSES

WILD-BARFIELD ELECTRIC FURNACES LIMITED

ELECFURN WORKS, OTTERSPOOL WAY, WATFORD BY-PASS, WATFORD, HERTS. Telephone: Watford 6091 (8 lines)

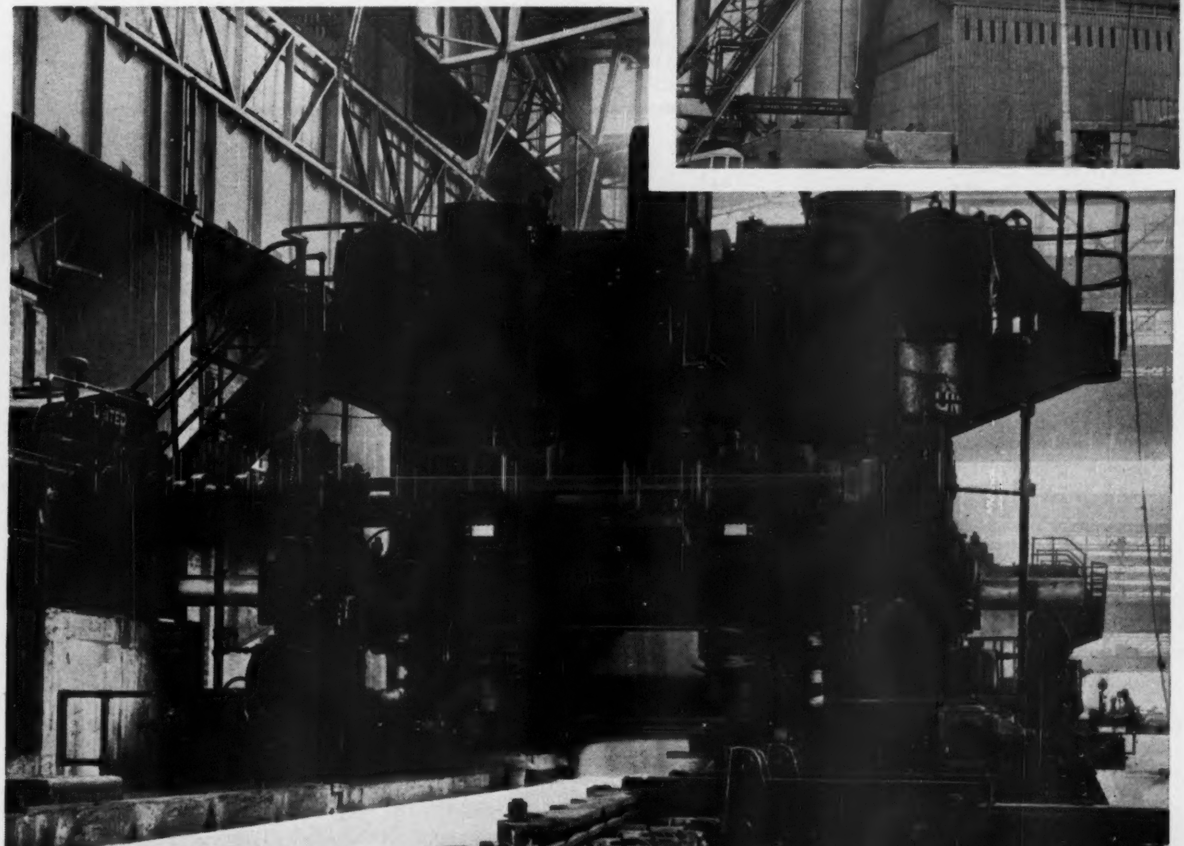
WB47

After leaving the soaking pits: an ingot being placed on the ingot buggy

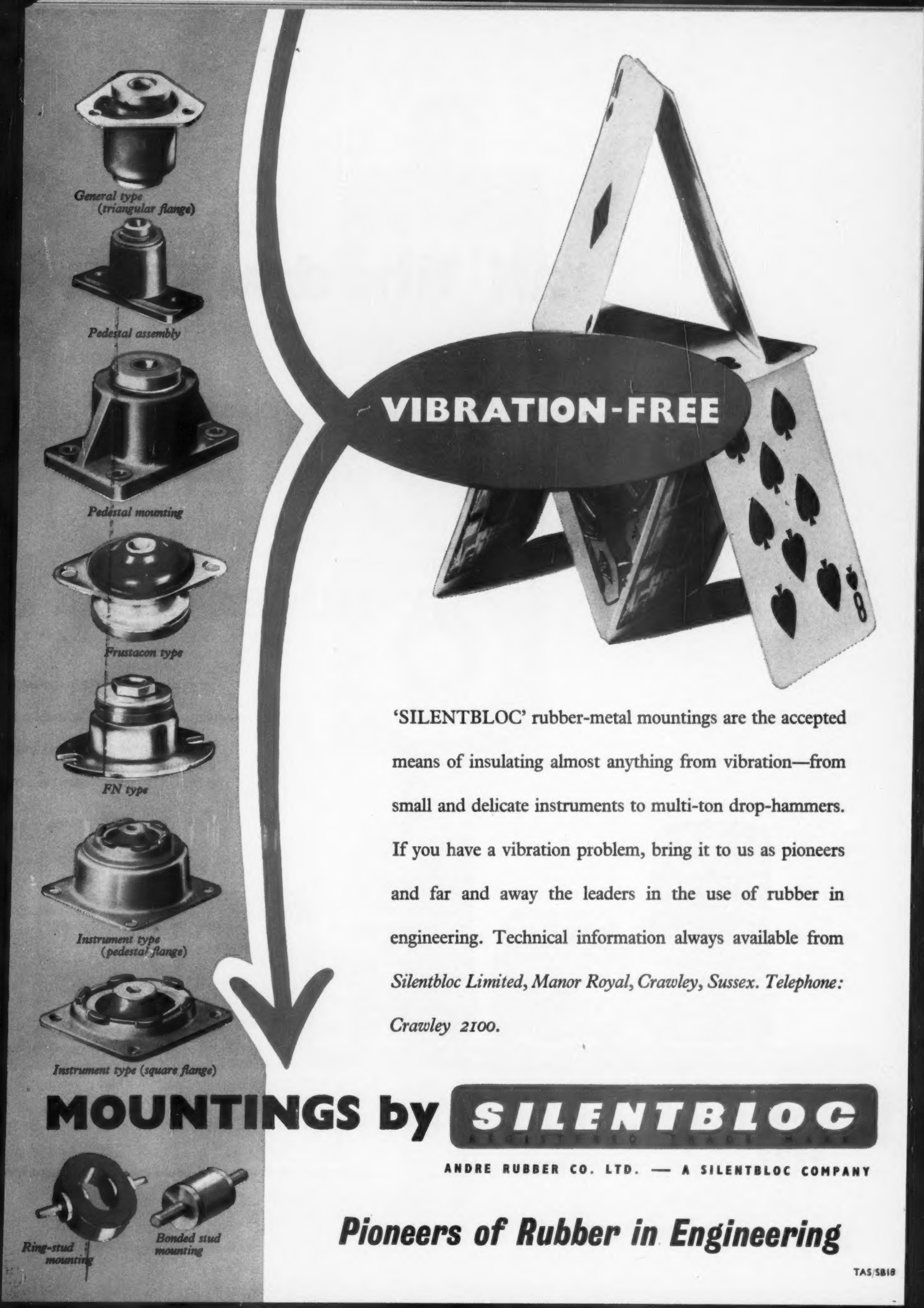


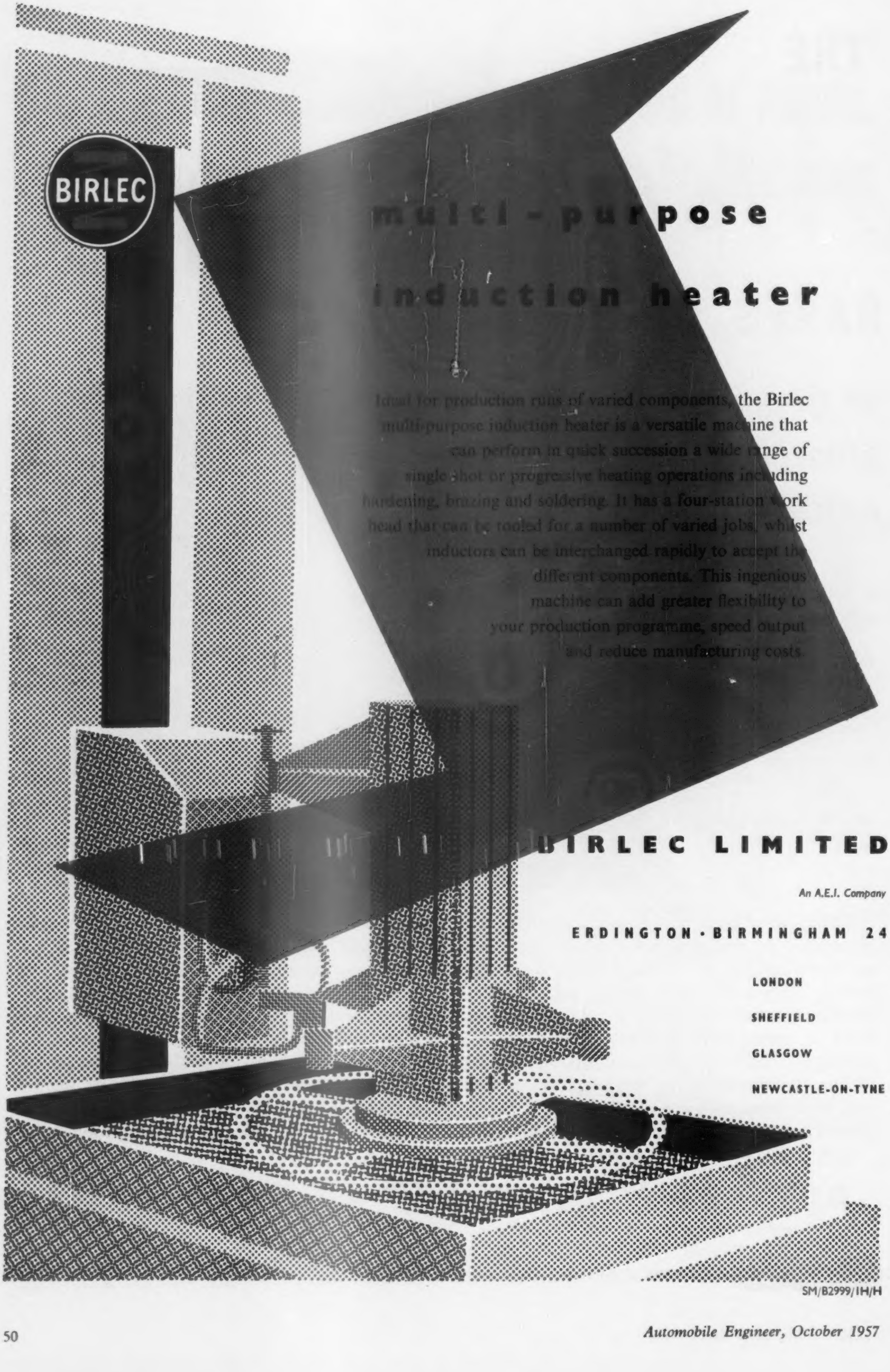
No. 4 Blast Furnace

A slab passing through one of the roughing stands of the 80" continuous hot-strip mill at Abbey Works




FASTER WELDING WITH
Argonarc
The aluminium body of this Rolls-Royce car is but one of many examples of the use of the British Oxygen Argonarc welding process in the motor industry. There are five good reasons why Argonarc is so extensively used. Its high welding speed results in little or no distortion; it produces sound, bright welds free from porosity, its gas shield eliminates the use of flux, so there are no post-weld corrosion problems; no expensive post-weld cleaning; an Argonarc welded joint is stronger than a riveted joint and, when dressed, is indistinguishable from the parent metal.
With acknowledgements to Park Ward Ltd.
BRITISH OXYGEN
British Oxygen Gases Limited, Industrial Division, Spencer House, 27 St. James's Place, London, S.W.1


General type (triangular flange)
Pedestal assembly
Pedestal mounting
Frustacon type
FN type
Instrument type (pedestal flange)
Instrument type (square flange)
VIBRATION-FREE
'SILENTBLOC' rubber-metal mountings are the accepted means of insulating almost anything from vibration—from small and delicate instruments to multi-ton drop-hammers. If you have a vibration problem, bring it to us as pioneers and far and away the leaders in the use of rubber in engineering. Technical information always available from Silentbloc Limited, Manor Royal, Crawley, Sussex. Telephone: Crawley 2100.
MOUNTINGS by SILENTBLOC
REGISTERED TRADE MARK
ANDRE RUBBER CO. LTD. — A SILENTBLOC COMPANY
Ring-stud mounting
Bonded stud mounting
Pioneers of Rubber in Engineering
TAS/SB18


BIRLEC
multi-purpose
induction heater
Ideal for production runs of varied components, the Birlec multi-purpose induction heater is a versatile machine that can perform in quick succession a wide range of single shot or progressive heating operations including hardening, brazing and soldering. It has a four-station work head that can be tooled for a number of varied jobs, whilst inductors can be interchanged rapidly to accept the different components. This ingenious machine can add greater flexibility to your production programme, speed output and reduce manufacturing costs.
BIRLEC LIMITED
An A.E.I. Company
ERDINGTON · BIRMINGHAM 24
LONDON
SHEFFIELD
GLASGOW
NEWCASTLE-ON-TYNE
SM/B2999/IH/H

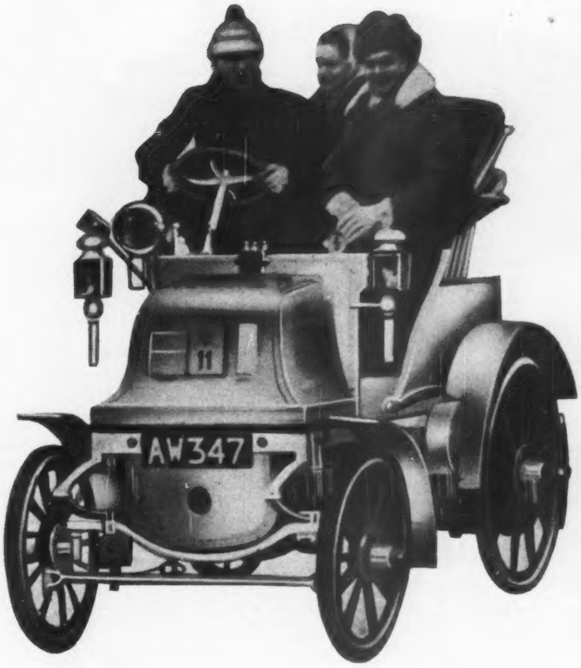
KIRKSTALL AXLES

WERE FITTED

TO THIS

1897

DAIMLER



THEY ARE USED TODAY
ON THE WORLD'S
BEST COACHES,
BUSES, TRUCKS,
AND DUMPERS, ETC.

KIRKSTALL FORGE ENGINEERING LIMITED

LEEDS, 5

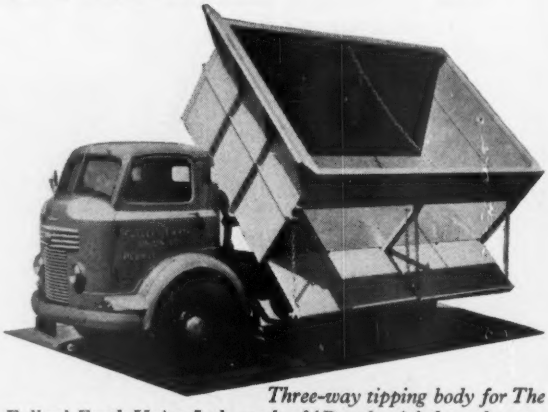
Telephone: Horsforth 2821

ENSURE PERMANENT FUEL SAVING WITH

DURALUMIN

REGD. TRADE MARK

Big reductions in tare and therefore in fuel consumption are made possible by using the strong, lightweight aluminium alloy 'Duralumin' in these vehicles. As strong as steel yet only one-third its weight, 'Duralumin' cuts running costs, reduces tyre wear, taxes and fuel consumption, and increases payload capacity. It does not rust and, if desired, may be left unpainted.
All the examples shown are the work of the Duramin Engineering Co. Ltd., one of the well-known body-builders producing designs in 'Duralumin'.

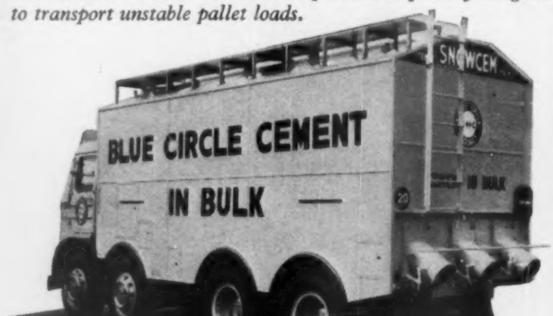


Three-way tipping body for The Fullers' Earth Union Ltd., made of 'Duralumin' throughout to save weight, withstand corrosion and hard wear.

Payload increase of 30 cwt. was made possible by using 'Duralumin' for the body and cab of this bulk flour transport—one of six made for Joseph Rank Ltd., carrying special equipment for delivering flour to a height up to 100 ft.

This special side-loader on an Albion chassis has dropsides, roller shutters and bodywork of 'Duralumin'. It is divided into compartments specially designed to transport unstable pallet loads.

The cab and thermostatically-controlled 7-ton capacity refrigerated body of this van are constructed from 'Duralumin', lined with hygienic 'MG5'. It will be operated in Ghana by the United Africa Company.



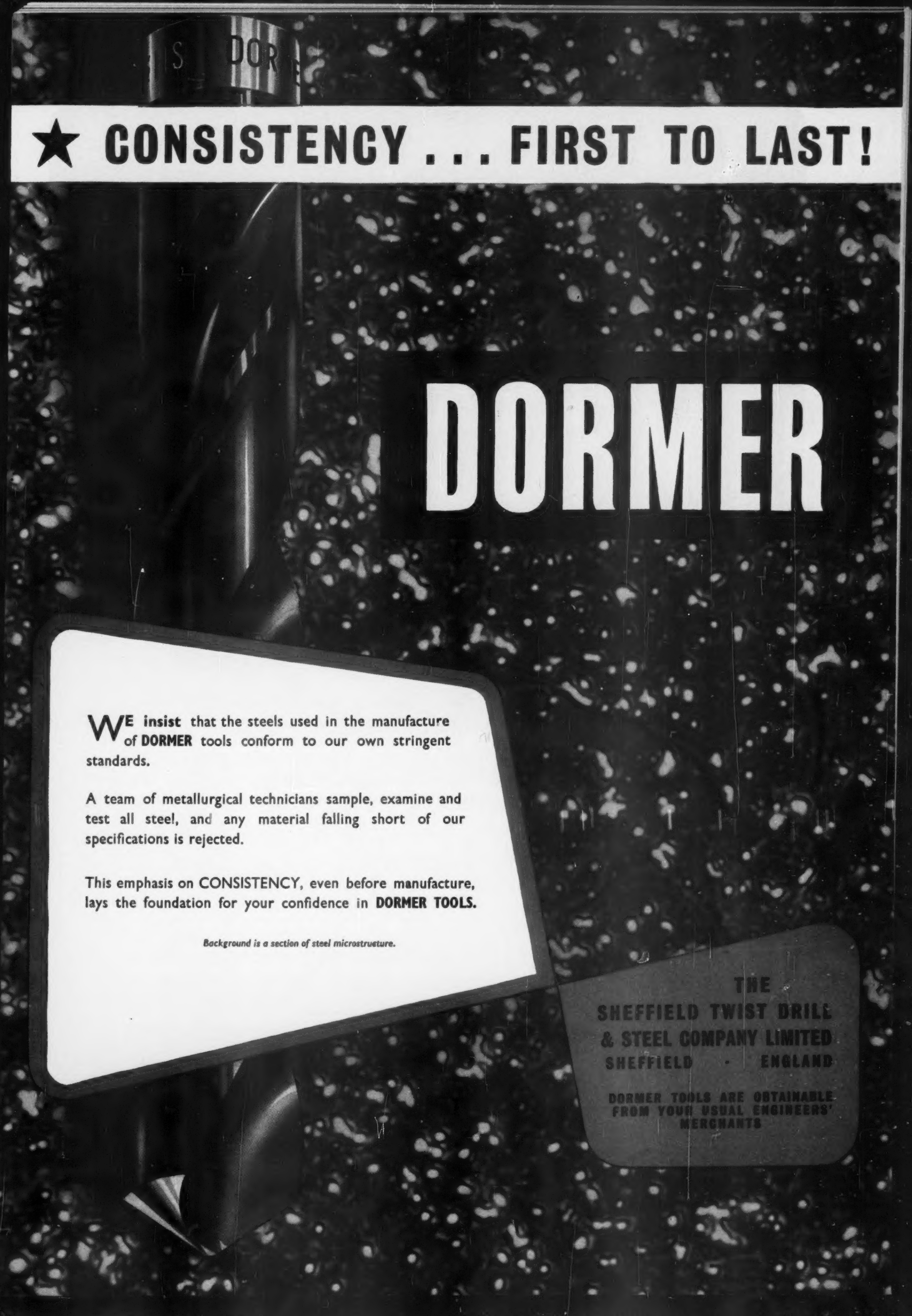
16-ton bulk cement light alloy body on 8-wheeler chassis, built for the Cement Marketing Company Ltd. It can be discharged satisfactorily at the rate of one ton per minute by the fluidisation system of aeration.

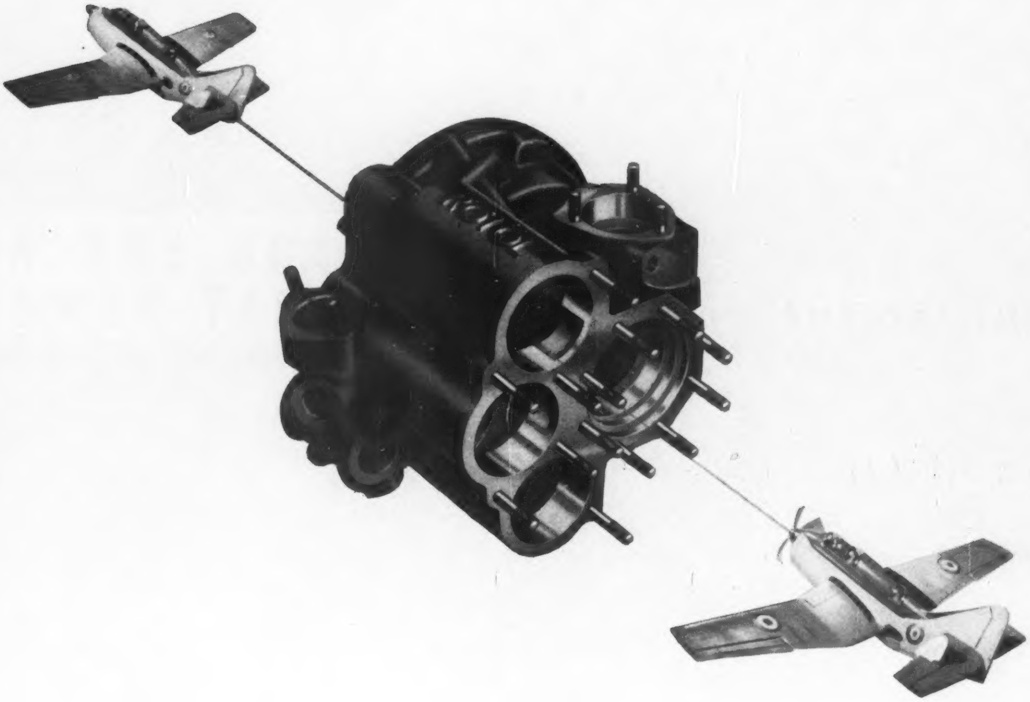
BOOTH **JAMES BOOTH & COMPANY LIMITED · ARGYLE STREET WORKS · BIRMINGHAM · 7**

Extrusions, large forgings, plate, sheet, strip, tubes and wire in brass and copper, as well as light alloys

TGA JBT 114


SDF
Smethwick Drop Forgings
...FITTED FOR GOOD!


★ CONSISTENCY . . . FIRST TO LAST!
DORMER
WE insist that the steels used in the manufacture of DORMER tools conform to our own stringent standards.
A team of metallurgical technicians sample, examine and test all steel, and any material falling short of our specifications is rejected.
This emphasis on CONSISTENCY, even before manufacture, lays the foundation for your confidence in DORMER TOOLS.
Background is a section of steel microstructure.
THE SHEFFIELD TWIST DRILL & STEEL COMPANY LIMITED
SHEFFIELD · ENGLAND
DORMER TOOLS ARE OBTAINABLE FROM YOUR USUAL ENGINEERS' MERCHANTS



ROTOL

CONFIDENTLY RELY ON P.B.M. QUALITY

Fairey Gannet anti-submarine aircraft are powered by the Armstrong Siddeley Double Mamba Engine which provides extremely powerful take-off and combat speed.

P.B.M. supply light alloy precision machined castings for the governor casings of the vital Rotol feathering u its used on these aircraft.

WHERE ACCURACY IS VITAL P.B.M. QUALITY IS SUPREME

For machined components the greatest names in aircraft engineering are listed with P.B.M. customers. Are you?

PBM

PERRY BARR METAL COMPANY LTD., GREAT BARR, BIRMINGHAM 22A

TELEPHONE: GREAT BARR 1794-5-6

ON THE SCAMMELL POWER TAKE-OFF

TRAILING-LINK FLEXIBLE couplings

The Metalastik Trailing Link Coupling is one of the most compact forms of coupling and occupies the minimum of axial length in relation to the parallel misalignment accommodated. It also takes care of angular misalignment and end float while cushioning shock loads and smoothing out torque irregularities. It has a high capacity for occasional overloads.

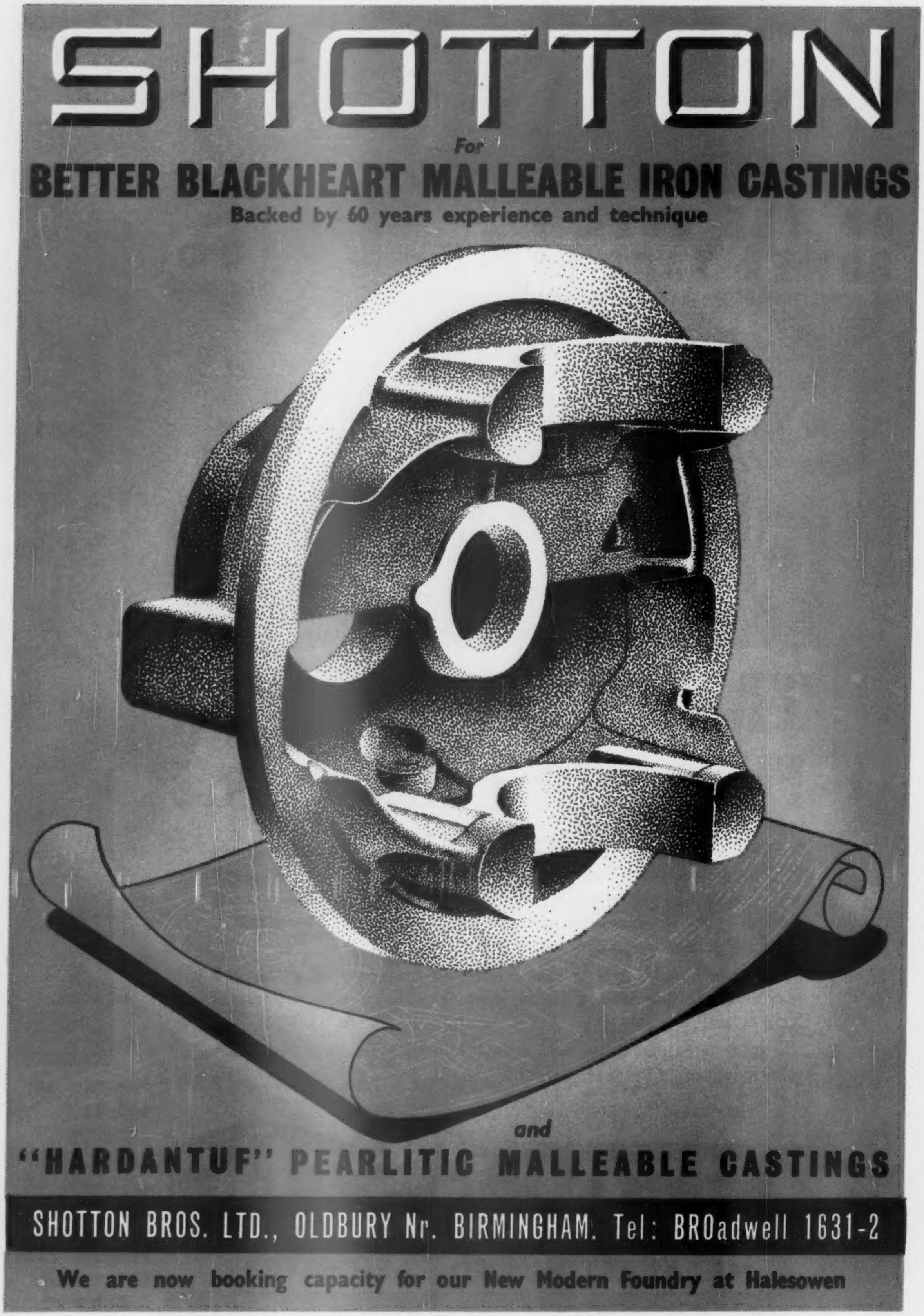
In the drive from gearbox to three-cylinder air-compressor on a Scammell road tanker, as illustrated, a Trailing Link Coupling transmits a torque of 2,250 lb./ins.

The small illustration shows the famous Metalastik Ultra-Duty Bush which takes movement in the links of the coupling.

For main as well as auxiliary drives, Trailing Link Couplings are manufactured in a range of sizes to transmit 1-20 h.p. at 100 r.p.m.

METALASTIK

METALASTIK LTD. LEICESTER


SHOTTON
For
BETTER BLACKHEART MALLEABLE IRON CASTINGS
Backed by 60 years experience and technique
and
"HARDANTUF" PEARLITIC MALLEABLE CASTINGS
SHOTTON BROS. LTD., OLDBURY Nr. BIRMINGHAM. Tel: BROadwell 1631-2
We are now booking capacity for our New Modern Foundry at Halesowen

Dismantling Expert

The old clock has seen better days. Now some of the parts are put to one side for future use. In the factory across the road, something similar is happening on an industrial scale.

The old machine shop has outlived its usefulness, so Thos. W. Ward Ltd. have been called in to demolish it. Those machines and materials which are still useful are being carefully preserved to form part of the great new machine shop which will rise on the site.

An undertaking of this kind in which speed is all important, calls for long experience and careful planning. Wherever dismantling and demolition are the prelude to industrial developments, Wards are the people to handle the contract.

Industrial dismantling is just one of the many activities of the Ward Group of Companies whose scope is as broad as industry itself.

SERVING INDUSTRY AROUND THE WORLD **THOS. W. WARD LTD**

GP/60

Head Office: ALBION WORKS · SHEFFIELD *London Office:* BRETTENHAM HOUSE · LANCASTER PLACE · STRAND W.C.2.

MARLES type 4 power-steering gear

This latest type 4 unit embodies the wide experience of Marles in power-steering, and consists of a 2-valve box mounted on the top of the standard Marles 861 manual gear unit.

The type 4 unit, the Universal unit and the standard manual unit are all interchangeable on the same mountings.

Further particulars will be sent on request.

ADAMANT ENGINEERING CO. LTD., DALLOW ROAD, LUTON

Sole proprietors of the Marles Steering Company Ltd.

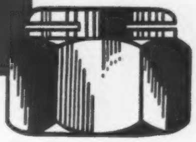
Telephone: Luton 2662 (4 lines) Telegrams: Adamant, Phone, Luton

MARLES

Photograph by courtesy of Ford Motor Co. Ltd.

FORD consistently rely on

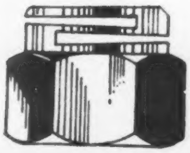
PHILIDAS Self-locking Nuts

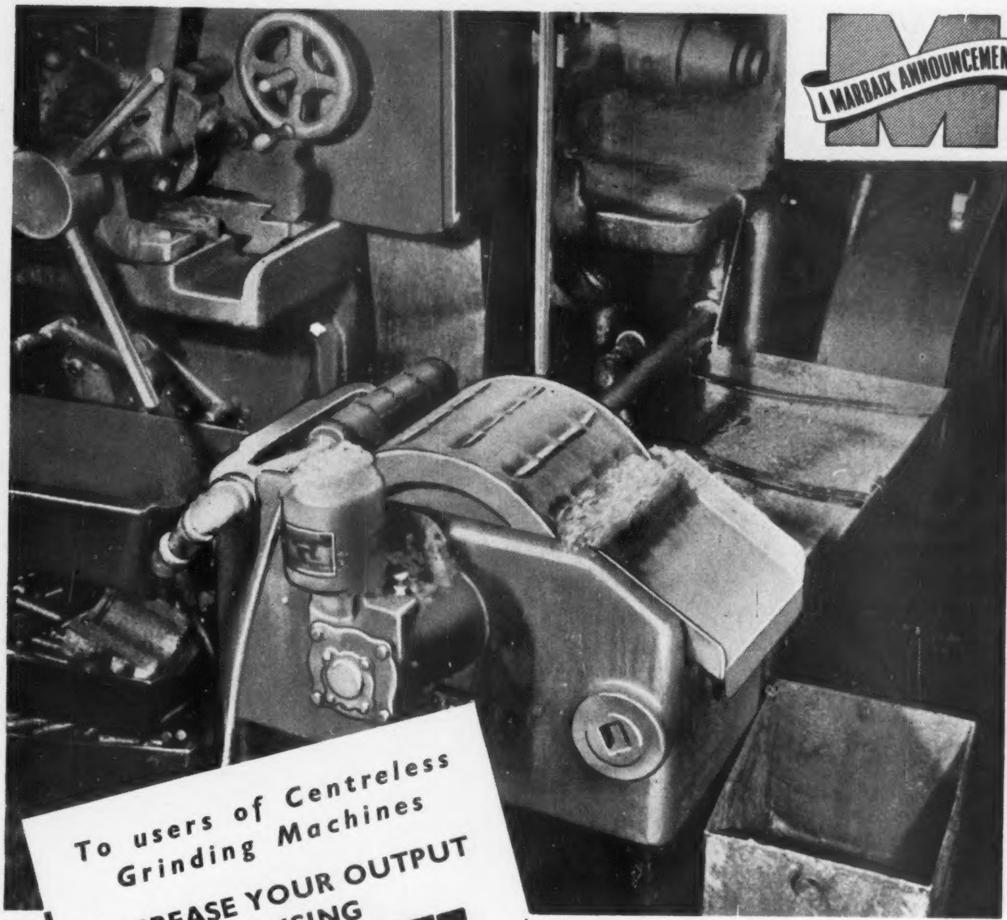


Under conditions of extreme temperatures or vibration these locknuts

YIELD ONLY TO A SPANNER

Self-locking for the Automobile, Aircraft, Electrical and Engineering Industries. Wheel Nuts, Anchor Nuts, Strip Nuts, etc.





A MARBAIX ANNOUNCEMENT

To users of Centreless Grinding Machines

INCREASE YOUR OUTPUT BY USING

BARNESDRIL

AUTOMATIC MAGNETIC COOLANT SEPARATORS

improve finish and reduce costs by removing ALL impurities

Increased output comes from fewer wheel dressings, easier wheel cutting, finer work finish and greater coolant efficiency. These benefits are supplied by the Separator AUTOMATICALLY through removal of all swarf and metal particles, and return of consistently clean coolant to the wheel.

Write to-day for illustrated catalogue.

Size No.	0,	1,	2,	4,	7,	10,
Capacity in imperial galls.	5,	10,	20,	41,	72,	103,

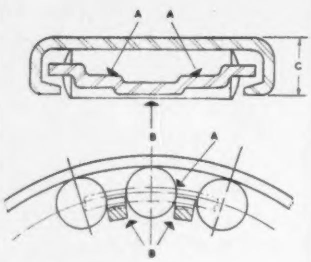
BRITISH MADE

PATENT Nos. 603083, 731655 AND OTHERS.

GASTON E. MARBAIX LTD

DEVONSHIRE HOUSE, VICARAGE CRESCENT,
BATTERSEA LONDON, S.W.11
PHONE BATTERSEA 8888 (8 lines)

Features of the new
TORRINGTON DRAWN CUP ROLLER BEARING



* Rollers end-guided at pitch line (A).
* Shaft-riding retainer (B) designed to permit lubricant circulation.
* High capacity in small cross section (C).
* Long pregreased life.
* Efficient at high speeds.
* Mounted by press fit.
* Simple housing design.
* Low unit cost.

Introducing

a new low-cost precision roller bearing . . .

THE TORRINGTON DRAWN CUP ROLLER BEARING

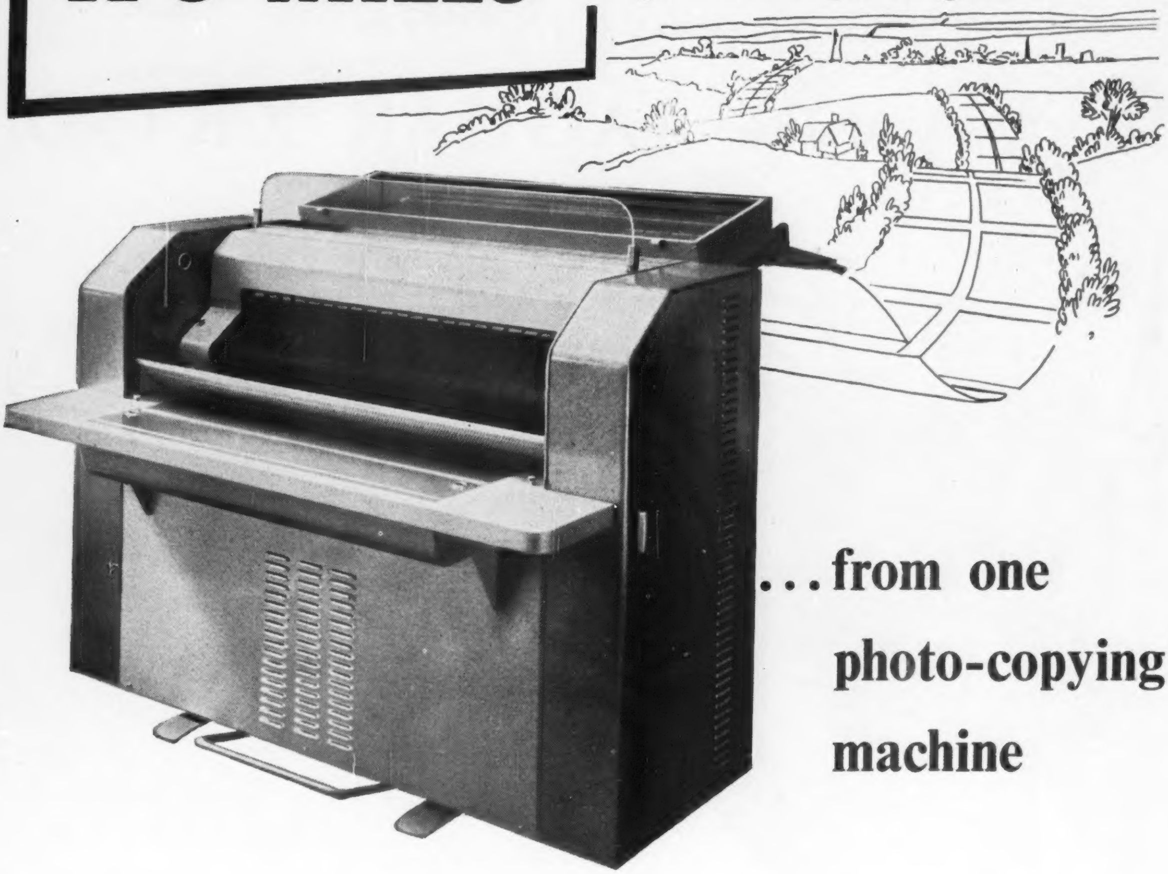
The advantages of drawn cup outer race construction are now available in a precision roller bearing. Designed to take up no more space than the Torrington needle bearing this compact, light-weight bearing consists of spherical end needle rollers, a one-piece hardened steel retainer and case-hardened thin-section outer race.
Highly efficient roller guidance and lubrication are outstanding features. The shaft-riding retainer contacts the roller ends at the pitch line where guidance can be obtained with the least effort. The design provides ample storage for lubricant and promotes its circulation.
These features make the new bearing particularly suited to applications requiring compactness with precision, high-speed endurance or long pregreased life. For further information of availability and prices write to our Bearings Division.

TORRINGTON BEARINGS

THE TORRINGTON COMPANY LTD. Works: Torrington Avenue, Coventry
London Office: 7-10 Eldon Street, EC2 Glasgow Office: 14 Moir Street, C1

817/1

175 MILES *of prints per year . . .*



. . . from one photo-copying machine

175 miles of prints a year is the output reported by one regular user of an Azoflex Model 46/35—and that's the sort of output that any busy drawing office can expect. The AZOFLEX photo-printing process combines several unique features. It's the *only* daylight reflex copying process. It's the *only* process to apply a measured quantity of developer irrespective of running speed, thus ensuring optimum quality. It avoids the use of dangerous chemicals and there are no unpleasant fumes. Operation is quiet and free from vibration. Exposing, developing and print delivery are all synchronised to ensure an even flow of finished work at all speeds.

AZOFLEX MODEL 46/35 combined synchronised printer and developer. Capacity: cut sheets and rolls up to 42 in. wide. Printing speed: up to 30 ft. per minute. Dimensions: Height 58 in. Depth 60 in. Width 72 in. Weight 1,100 lb. Price £1,555 (including delivery and installation in the U.K.).

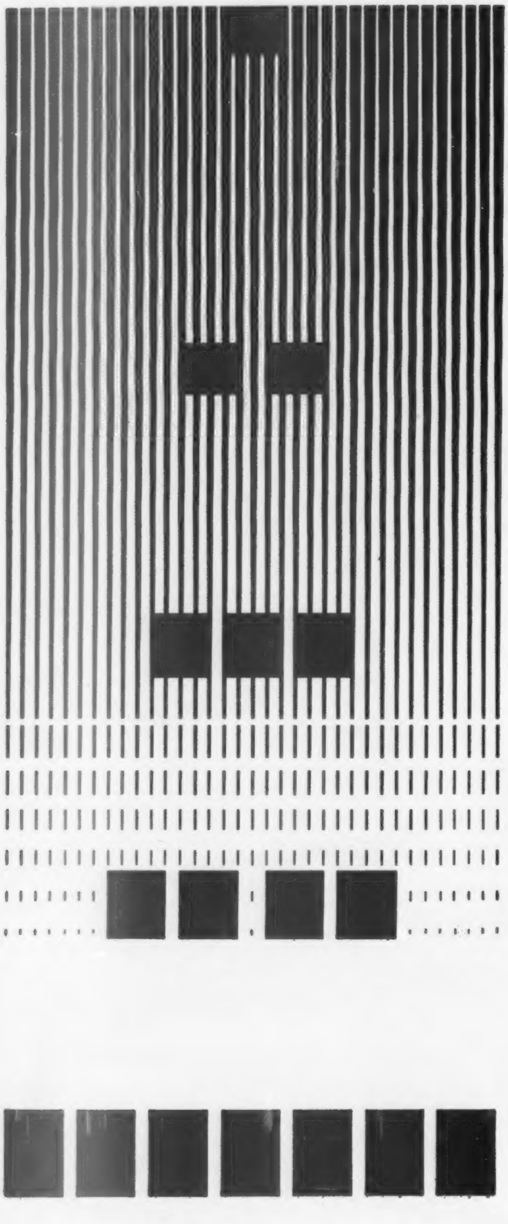
ILFORD *Azoflex*

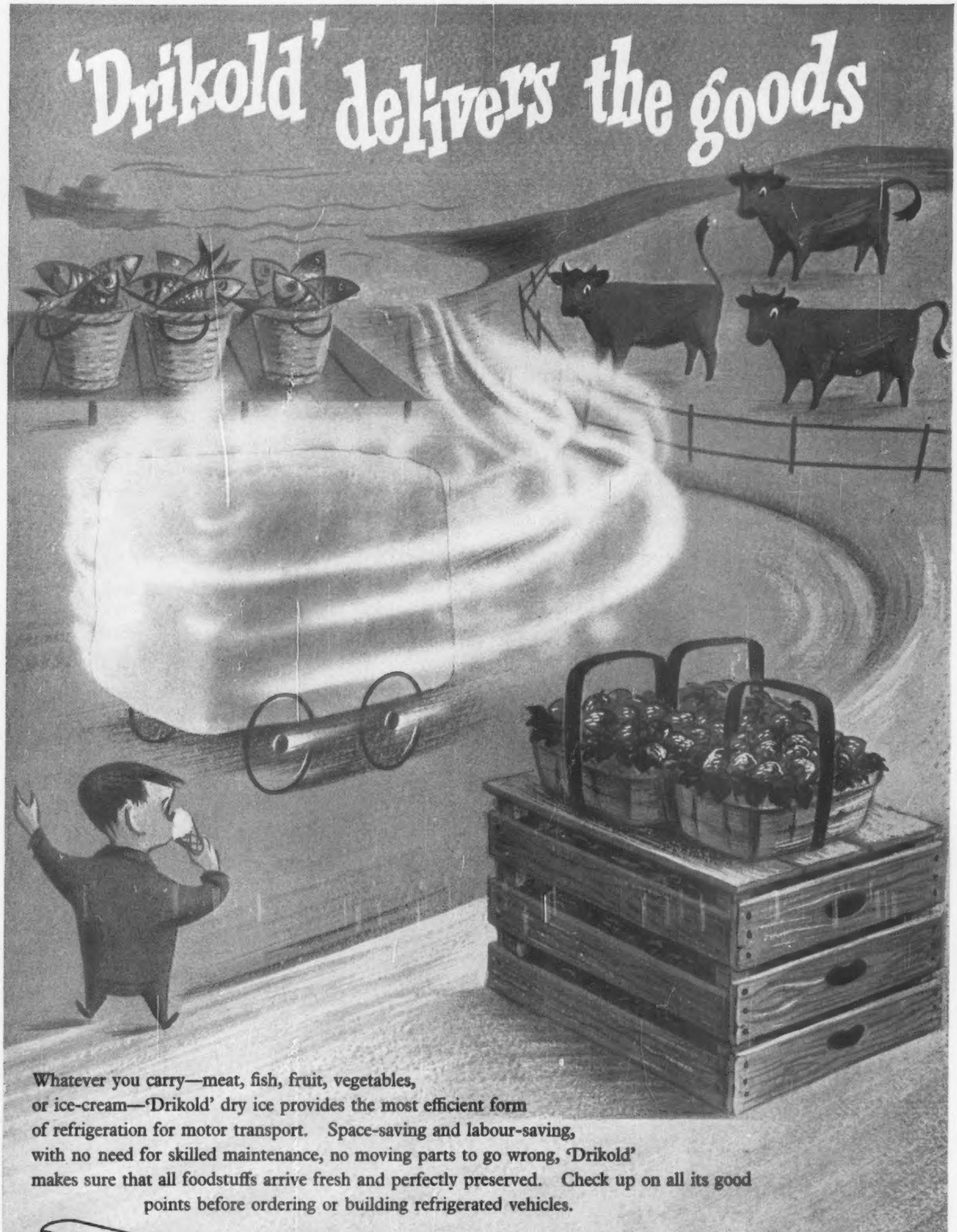
Enquiries to: Ilford Limited, Azoflex Dept. AZ11B
104 High Holborn, London, W.C.1. Telephone: HOLborn 3401

PHOTO-PRINTING MACHINES AND MATERIALS


COLES
THE NAME THAT CARRIES WEIGHT
Speed, f.p.m. ... 50
DERRICKING—Maximum to minimum
SLEWING—2 r.p.m. with load.
TRAVELLING—4 m.p.h. unloaded and 2 m.p.h. loaded on
Turning Radius—20 ft.
Ground Clearance—11 ins.
Weight—With standard jib 15 tons approximately.
MODEL S710C ARGUS
S.T. These people can offer the largest range of mobile cranes in the country and cater for special industries. Call in their representative and see what he can do for us.
F. Thompson
Designed, manufactured and marketed by :-
STEELS ENGINEERING PRODUCTS LIMITED
Sunderland, England Tel: 56281 (10 lines) Grams: Steel, Sunderland
SALES AND SERVICE: Glasgow: 235 Bath St., C.2 · Newcastle: Brunswick House, Brunswick Place
London: 143 Sloane St., S.W.1 · Manchester: 153 Oxford Road, 13 · Birmingham: 39 Thorp St., 5

246




'Drikold' delivers the goods
Whatever you carry—meat, fish, fruit, vegetables, or ice-cream—'Drikold' dry ice provides the most efficient form of refrigeration for motor transport. Space-saving and labour-saving, with no need for skilled maintenance, no moving parts to go wrong, 'Drikold' makes sure that all foodstuffs arrive fresh and perfectly preserved. Check up on all its good points before ordering or building refrigerated vehicles.
Food's all the fresher with 'Drikold'
For full details, please get in touch with
Imperial Chemical Industries Ltd., London, S.W.1.
ICI

DK.253

Don't buy oil seals haphazardly. Get Pioneer Oil Seals.
Pioneer representatives are factory trained men who can deal with any oil-sealing problem on the spot and recommend the seal you need.
Pioneer Personal Oil Sealing Service means not only studying a customer's individual needs; it also means recommending the right oil seals and personal inspection when they are in use.
Buy Pioneer and make sure your oil-sealing problems are solved for all time.

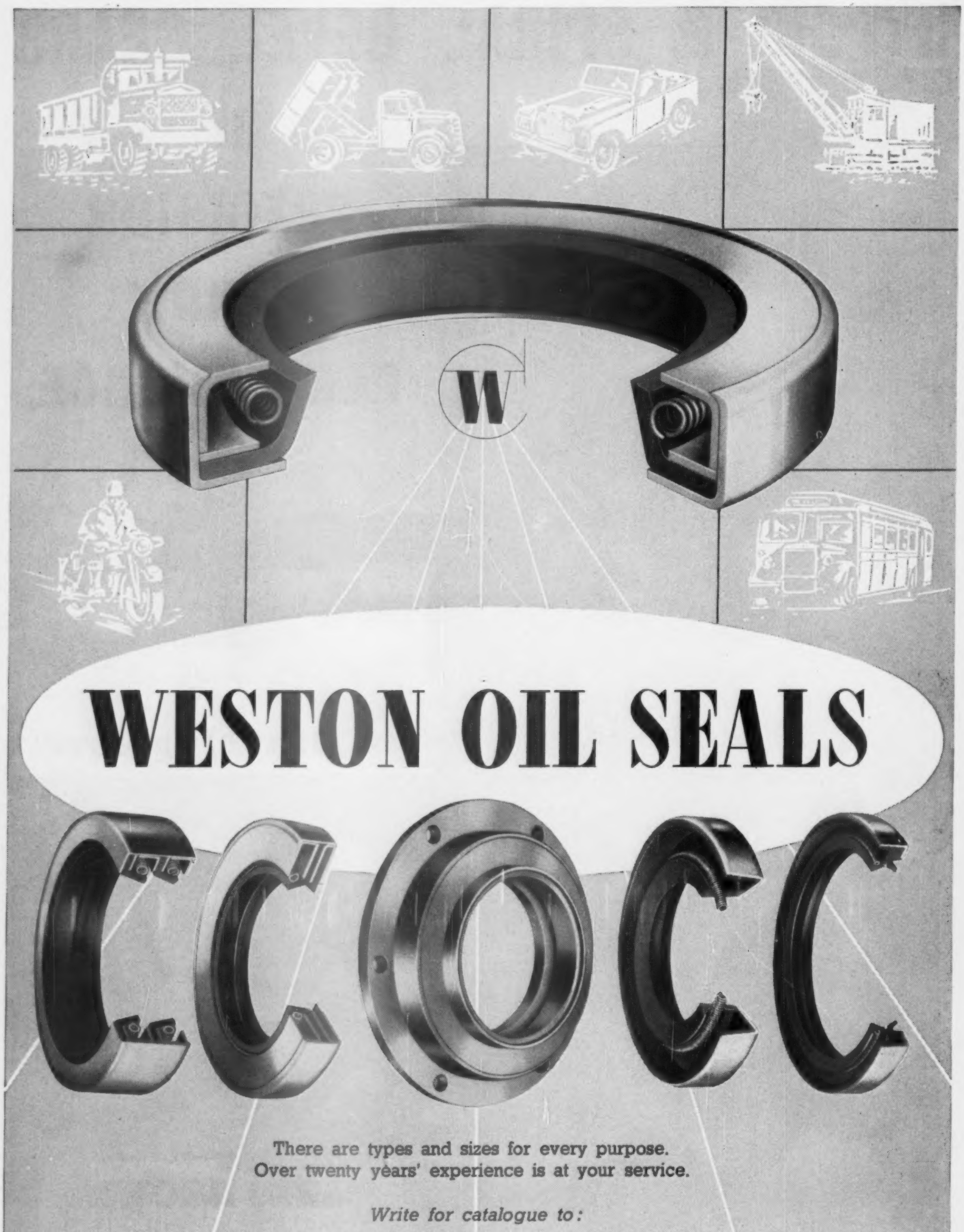
Write now for the most comprehensive catalogues in the trade on Oil Seals and O-Rings and for your copy of the OIL SEAL JOURNAL

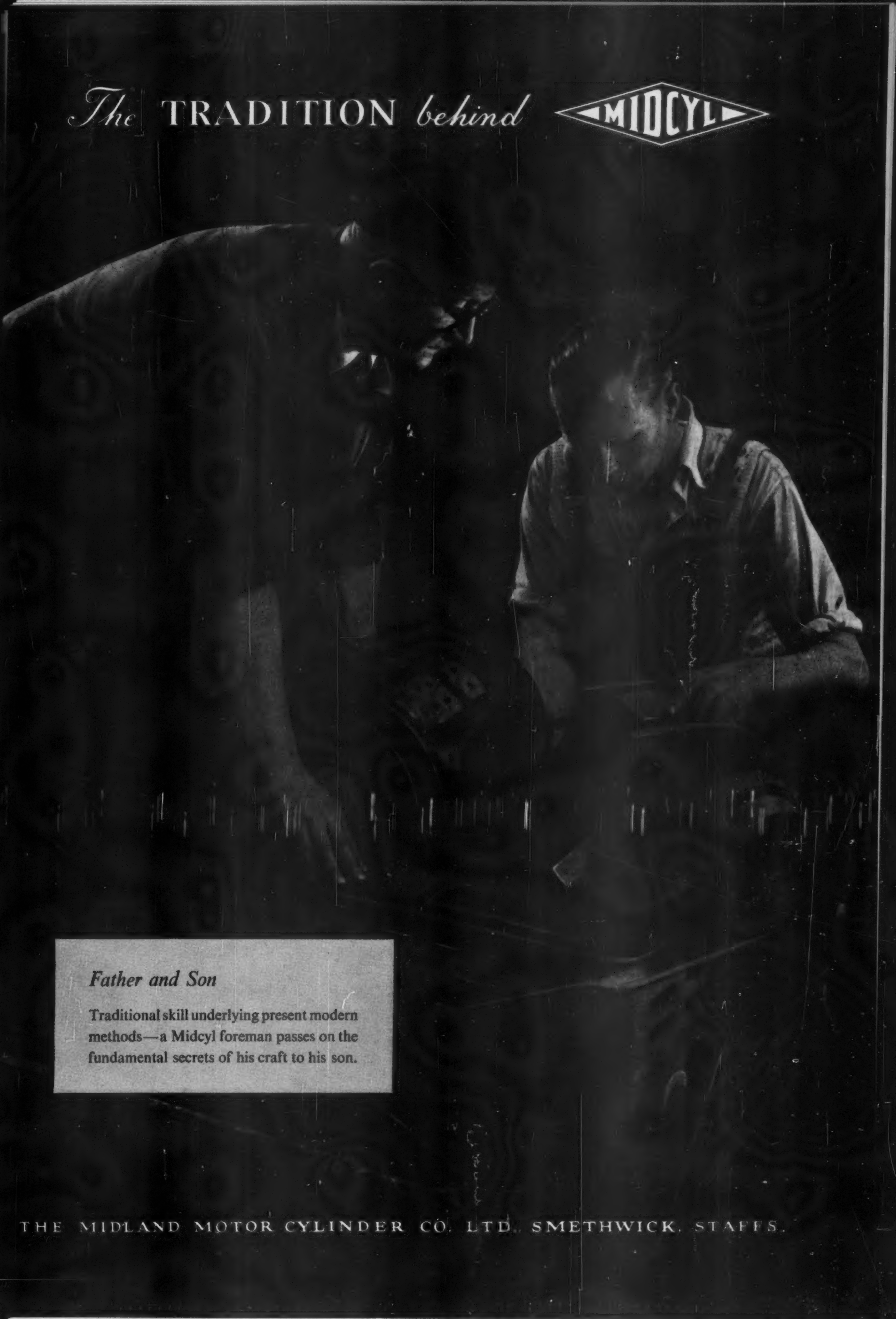
PIONEER

Oilsealing & Moulding Co. Ltd.

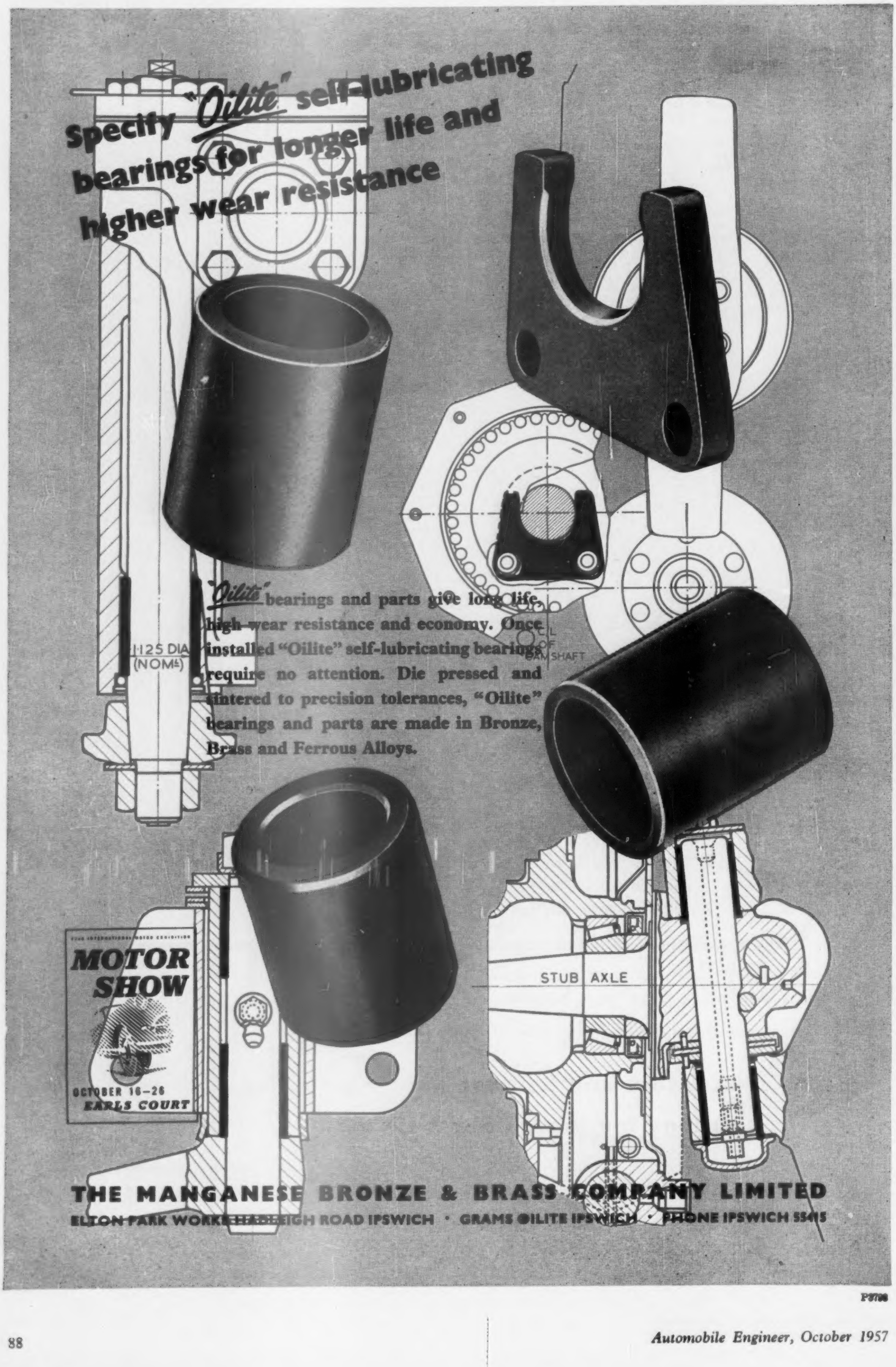
A DIVISION OF J. H. FENNER & CO. LTD.

Factory & Head Office: Cottontree Works, Colne, Lancashire · Telephone: Wycoller 411/2/3


W
WESTON OIL SEALS
There are types and sizes for every purpose.
Over twenty years' experience is at your service.
Write for catalogue to:
CHARLES WESTON & CO. LIMITED
Irwell Bank Works · Douglas Green · Pendleton · Salford 6.
Telephone: Pendleton 2857-8-9 Birmingham: Midland 6952
London: Holborn 0414.


The TRADITION behind MIDCYL
Father and Son
Traditional skill underlying present modern methods—a Midcyl foreman passes on the fundamental secrets of his craft to his son.
THE MIDLAND MOTOR CYLINDER CO. LTD. SMETHWICK. STAFFS.

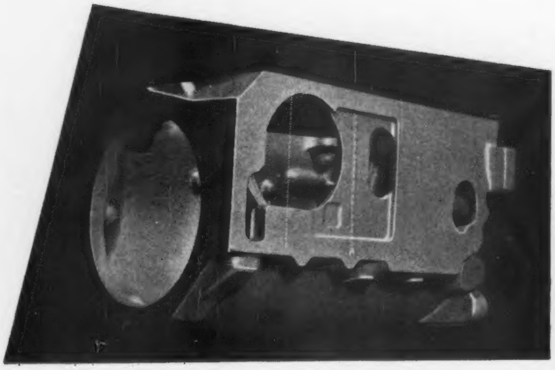

GIRLING DISC BRAKES
ARE MAKING A WORLD OF DIFFERENCE
See the latest developments on
Stand 318
THE BEST BRAKES IN THE WORLD
GIRLING LIMITED · KINGS ROAD · TYSELEY · BIRMINGHAM 11


Specify "Oilite" self-lubricating bearings for longer life and higher wear resistance
"Oilite" bearings and parts give long life, high wear resistance and economy. Once installed "Oilite" self-lubricating bearings require no attention. Die pressed and sintered to precision tolerances, "Oilite" bearings and parts are made in Bronze, Brass and Ferrous Alloys.
1·125 DIA (NOML)
C/L OF CAM SHAFT
STUB AXLE
MOTOR SHOW
OCTOBER 16–26
EARLS COURT
THE MANGANESE BRONZE & BRASS COMPANY LIMITED
ELTON PARK WORKS HADLEIGH ROAD IPSWICH · GRAMS OILITE IPSWICH · PHONE IPSWICH 55415

P3798

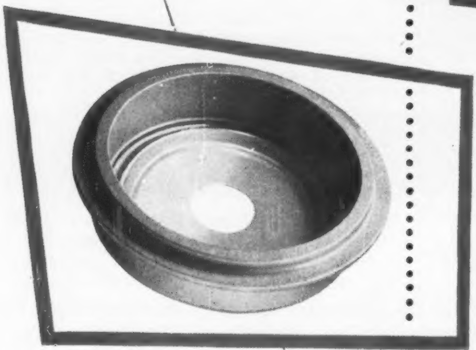
Darcast keeps pace with Progress

The Control and Technique
developed for the production of
High Duty Iron Castings
by the Darcast Foundries
is ready to meet developments
of the Motor Industry.



AUTOMATIC GEARBOX CASE *(COURTESY OF ROLLS-ROYCE LTD)*

BRAKE DISC *(COURTESY OF DUNLOP RIM & WHEEL CO LTD)*



COMPOSITE STEEL & CAST IRON BRAKE DRUM
(COURTESY OF VAUXHALL MOTORS LTD)

DARCAST

DARTMOUTH AUTO CASTINGS LIMITED
SPECIALISTS IN HIGH DUTY IRON CASTINGS FOR THE MOTOR INDUSTRY

SMETHWICK 40
BIRMINGHAM

AUTOMOBILE ENGINEER

CONTENTS

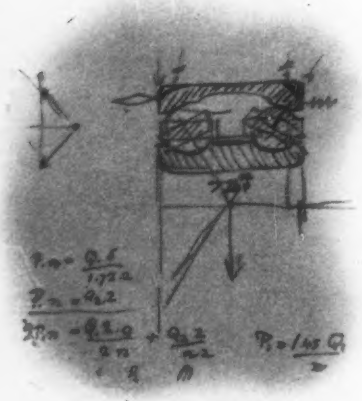
EVEN SUCH RELATIVELY LARGE AND EASILY DEFORMABLE COMPONENTS AS THIS ELECTRICAL INSTRUMENT CHASSIS IN PRESSED STEEL SHEET CAN BE DEBURRED AND POLISHED EFFICIENTLY IN MODERN BARRELLING EQUIPMENT

Published the second Wednesday in every month
by ILIFFE & SONS LIMITED
*Dorset House, Stamford Street, London, S.E.*1
Telephone · Waterloo 3333 *(60 lines)*
Telegrams · Sliderule, Sedist London
The annual subscription inland and overseas is £3 0s 0d *including the special number*
Canada and U.S.A. $8.50

BRANCH OFFICES
Coventry · 8-10 *Corporation Street*
Telephone · Coventry 5210
Birmingham · King Edward House, New Street, 2
Telephone · Midland 7191
Manchester · 260 *Deansgate,* 3
Telephone · Blackfriars 4412 *and Deansgate* 3595
Glasgow · 26B *Renfield Street, C.*2
Telephone · Central 1265

VOLUME 47 NUMBER 10 OCTOBER 1957

The Conquest of Friction



In 1907, Dr. Sven Wingquist, a young engineer in a textile mill, and inspired no doubt by alignment difficulties of the line shafting then so much in vogue for power transmission, worked out the basic principles of a new type of bearing, the self-aligning ball bearing.

This marked a major advance in anti-friction engineering; from this inspired beginning has grown the whole world-wide SKF organisation.

THE SKEFKO BALL BEARING COMPANY LIMITED · LUTON · BEDS

THE ONLY BRITISH MANUFACTURER OF ALL FOUR BASIC BEARING TYPES: BALL, CYLINDRICAL ROLLER, TAPER ROLLER AND SPHERICAL ROLLER

SKF

G156

Editor *J. B. Duncan*

Editorial Staff *T. K. Garrett, A.M.I.Mech.E. A.F.R.Ae.S.(Associate Editor)*

F. C. Sheffield

DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Export Markets

UNLESS there are unexpected changes in the British economy, there is little prospect that artificial restrictions such as purchase tax and onerous hire purchase terms will be lifted to make the home market for road vehicles more buoyant. Therefore, in order that adequate use can be made of the increasing production facilities it will be necessary to continue the export drive in an even more energetic manner.

The creation of an export demand, and its future retention and expansion, raise certain problems differing, some only in degree and others in kind, from those of the home market. Providing the vehicle is basically satisfactory, the first requisite for the export market is that spares and replacement parts shall be quickly available. There was a time when the British record in this respect was not too high—fortunately that is no longer true. The Rootes Group, for instance, some months ago inaugurated an air service for spares to Central Africa, and they intend to extend this type of service to other parts of the world.

BMC are also taking effective steps to deal with this problem. In the United States of America, they, in conjunction with Hambro Automotive Corporation, have established a seven-point service plan, which includes:

1. Large fully stocked parts depots in New York and San Francisco.
2. Substantial distributor parts inventories in twelve major cities to supplement the parts stocks of 500 dealers.
3. Service schools to be attended by general service managers of all BMC distributors in the country.
4. Additional mechanics' service schools for the benefit of dealer service managers and mechanics.
5. Adequate technical service assistance from Hambro central service depot.
6. Service guide manuals outlining minimum requirements on all service matters.
7. Improved after-sales-service.

The importance of after-sales service, with quickly supplied and reasonably priced replacement parts, cannot be too highly stressed; in itself it will not create a market, but help in its extension. Export markets can be created only if the vehicle itself is fundamentally sound, and is suitable, perhaps with slight modifications, for service under widely differing conditions. The success of the export drive has, of course, raised problems, since British cars are exported to countries varying from sub-arctic to tropical. Obviously, a vehicle designed specifically for sub-arctic operation would differ considerably from one designed specially for tropical service. It is manifestly impossible to design for only one set of conditions, but fortunately, it is possible to produce vehicles which with, at the most, minor modifications, will function well under widely differing conditions of temperature and humidity.

To consider only one point, comfort for both driver and passengers. Ride comfort is, of course, mainly a matter of suspension, but for all-round comfort the temperature inside the body of the car is also important. So far as cold weather is concerned, this is no great problem; in practically all modern cars, provision is made for the installation of a heater, if one is not included as standard.

It is a very different matter to maintain a comfortable temperature for the occupants of a vehicle in a very hot climate. This calls for air-conditioning plant, and at present only Rolls-Royce of British cars can be obtained with such equipment. At first sight it may seem that air-conditioning equipment would be too expensive for installation in low and medium price cars. Admittedly the cost would be high in relation to the cost of the car, but American experience has shown that there is a demand of reasonable size for cheaper cars with air-conditioning units.

At least one organization in this country is carrying out extensive development work on this question. There is no doubt that the unit will perform the function for which it is intended, but its adoption will call for co-operation from the vehicle manufacturers. For optimum functioning, the best location for the unit would be under the bonnet in the engine compartment—this might call for some modification of the bonnet form. The alternative would be to locate the unit in the boot. This is not as good functionally, and has the added disadvantage that much of the boot space would be taken up, which could be a serious drawback in view of the fact that in the countries where air-conditioning is desirable, long journeys are common and much luggage is carried. It would be a feather in the cap of the industry if the first medium size passenger car to have complete air-conditioning equipment were to come from a British factory.

The B.S.A. Superchargers

A Range of Light, Compact Units, Suitable for Automotive and Other Diesel Engines

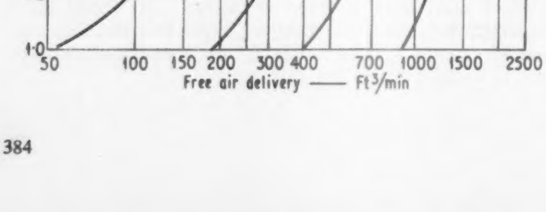
THE B.S.A. group of companies has been associated with the development of components for gas turbine engines since the earliest days of these power units. Much of their work in this connection has included the formulation of heat-resistant steels. In addition, they produce a wide variety of precision castings and components, such as turbine and compressor blades and discs. By virtue of this background of experience they have been well placed to develop their range of efficient and compact turbochargers. Moreover, by applying the techniques arising from modern developments in the group's manufacturing processes, they have been able to produce these turbochargers at a remarkably low cost.

So far, two sizes of turbocharger are ready for production. One is for diesel engines capable of developing from 200 to 400 b.h.p; the other, now being produced in batch quantities, is a smaller unit suitable for engines up to 200 b.h.p. This latter unit weighs 30 lb and its overall dimensions are approximately 8½ in diameter × 8 in long. Its maximum continuous rating is at a 2 : 1 pressure ratio and an exhaust temperature of 650 deg C. Trials have shown that an 8·6 litre engine equipped with this unit can be used to replace a 10·6 litre naturally aspirated engine with resultant fuel economies of as much as 10 per cent. The larger of the two turbochargers, which weighs 45 lb, is available for engine trials.

The basic principle of turbocharging, of course, is well known. Briefly, the exhaust gases from the engine are used to drive a turbine wheel, which, in turn, drives a compressor to boost the air supply to the engine. Since the amount of fuel that can be burned in a given time in the engine cylinders is determined by the amount of air that the unit can inhale, compression of the air before it enters the cylinders leads to the development of greater power outputs than can be obtained by natural aspiration.

Experience indicates that the majority of naturally aspirated engines currently in production can be boosted to 1½ to 2 atmospheres without modification. Even if modifications are necessary, for example, to meet requirements for higher boosts, they generally affect only the exhaust valves and piston assemblies, since bearing loadings are not greatly increased by supercharging. In any case, with very high-speed engines, the design criterion for bearings may be inertia rather than gas loading. There is no reason why engines designed specifically for turbocharging should not operate at boosts appreciably higher than 2 atmospheres.

Fig. 1. Engine capacities catered for by the B.S.A. turbochargers



Principal advantages

Since turbochargers generally are relatively small and light, the installed weight of turbocharged engines is far less than that of the naturally aspirated units giving the same power output. Similarly, an appreciable saving in installation space is possible, and the strength and weight of the frame required to carry the engine is also reduced. Moreover, an engine fitted with a turbocharger of economical design costs appreciably less than a naturally aspirated unit of the same power output.

Because of the improved overall efficiency of a turbocharged engine, the specific fuel consumption is lower than that of a naturally aspirated unit of the same power output. Two factors lead to this improvement in efficiency. One is that the mechanical losses in the turbocharged engine are not increased proportionately with the increase in power obtained by boosting the naturally aspirated version, while the second is that exhaust energy that otherwise would be wasted is used to boost the air flow to the cylinders, and this reduces the pumping losses of the engine.

The characteristic of the turbocharger can be matched to that of the engine to give the torque curve most suitable for the service for which the engine is to be employed. These requirements may be on the one hand for engines running continuously at maximum power output, or on the other hand for those requiring the highest torque output at low engine speeds. Only a very efficient turbocharger will meet the latter requirement satisfactorily.

Since, in operation, the speed of the turbocharger depends on the velocity of the exhaust gases passing through the turbine, the degree of turbocharging effected tends normally to increase automatically with engine speed and load. The overall noise level of a turbocharged engine generally can be reduced to a lower level than that of the naturally aspirated equivalent, because the exhaust gases, although entering the turbine as a series of high energy impulses, leave at a uniform velocity and a much reduced energy level. Similarly, the compressor smooths out the impulses caused by the opening and closing of the valves of the individual cylinders of the engine.

In general, the advantages of the turbocharged engine can be summarized as follows. Power output is increased in proportion to the degree of turbocharging. The overall power : weight ratio is considerably improved, and the installation space requirement for an engine of given output is reduced. Specific fuel consumption is reduced over a wide part of the operating range, and the initial cost of the engine per unit of power output is less than that of a naturally aspirated engine. Also, better torque-speed characteristics are obtainable. Another advantage, which is important for certain overseas markets, is that the power output obtained at sea level can be maintained at high altitudes. With the

B.S.A. turbocharger, because of the light weight of its rotating parts, there is an almost immediate response of the engine to changes of speed and load.

Economics of turbocharging

For a given output, the size of an engine varies inversely as the charge density; for instance, a six-cylinder engine turbocharged at a pressure ratio of 1·6 : 1 could, in certain applications, replace a naturally aspirated, eight-cylinder engine of the same bore and stroke. An even greater economy in size could be effected if the air charge, in passing between the turbocharger and the engine, were cooled to normal atmospheric temperature. Since the engine does not have to be altered structurally when a turbocharger is installed, its weight and size, and cost are not increased, so the additional cost is solely that of the turbocharger and its installation and the modified manifolds.

Quantitatively, the gains obtainable are indicated by the following data. A certain 200 b.h.p, naturally aspirated, high-speed diesel engine can be made to produce 270 b.h.p. simply by incorporating a turbocharger unit weighing only 30 lb. In the naturally aspirated condition, its specific weight is 12 lb/b.h.p, whereas in the turbocharged condition its specific weight is only 9 lb/b.h.p. So far as the engine alone is concerned, if its specific cost is £5 per b.h.p. in the naturally aspirated condition, its specific cost when turbocharged will be reduced to approximately £4 per b.h.p. As has already been mentioned, the economies obtained with regard to the frame and installation must also be taken into account when choosing between a turbocharged and a naturally aspirated engine.

The degree of fuel economy attained depends on the type of service for which the engine is used. Owing to its inherent flexibility, the B.S.A. turbocharger can be matched to give the best fuel consumption under the conditions in which the engine is mainly used. At speeds other than idling, this generally gives an improvement of between 5 and 10 per cent in specific consumption; between these limits, the figure actually obtained depends on the relative weight given to factors other than fuel economy, when the matching is carried out. Turbocharging should not have any adverse effects on maintenance requirements; in fact, for the larger size engines, the opposite has been claimed to have been experienced.

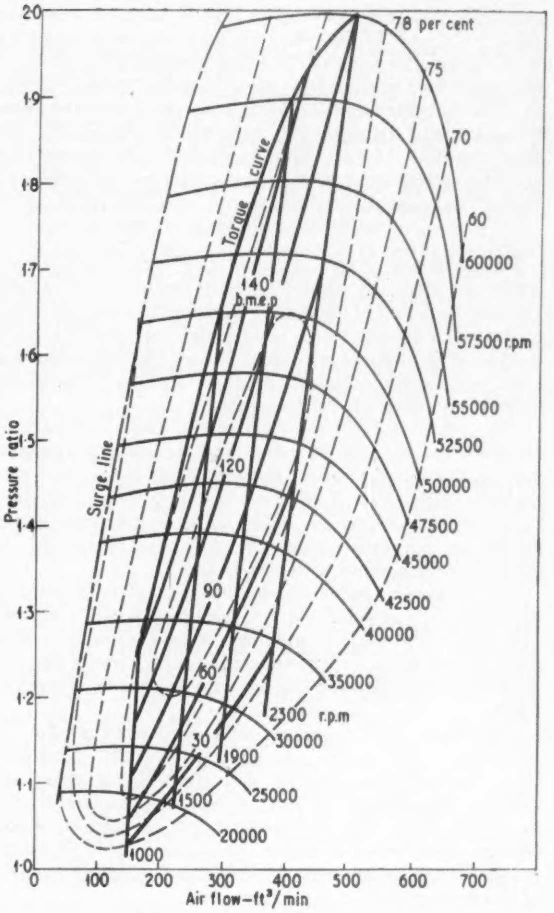


Fig. 2. The matching of the engine and compressor air flow characteristics, which are shown in heavy and light lines respectively

Matching the turbocharger to a diesel engine

The range of air flows obtainable with B.S.A. turbocharger units is shown in Fig. 1. From this, it can be seen that there should be no difficulty in selecting a turbocharger for any engine, provided its requirements are within the range of 50 to 2,000 ft³/min. It will probably be necessary, of course, to modify the exhaust and manifold porting to suit the turbocharger. The fuel injection equipment will also have to be adjusted to give the increased output, but it is important to avoid unduly lengthening the injection period. Apart from these, no other alterations should be necessary.

At first sight, it might appear to be surprising that the majority of engines currently in production can be supercharged to 1½ to 2 atmospheres, without major modifications. However, there are several reasons for this. One is that the maximum cylinder pressure can be controlled, within limits, by changes in injection timing or compression ratio. Secondly, the higher temperature and density of the charge at the end of the compression stroke increase the rate of heat transfer to the fuel droplets injected into the cylinders: consequently, the ignition delay and the high rate of pressure rise associated with it are reduced. This tends to reduce the shock loading on the bearings and is also one of the factors that helps to make turbocharged engines quieter than naturally aspirated units. A further advantage associated with this improvement in combustion characteristics is that lower quality fuels can be employed.

Turbocharging generally increases the maximum cylinder pressure about 200 lb/in² above that of naturally aspirated engines, which is commonly in the order of 1,100 lb/in². Despite this relatively small increase in maximum pressure, the b.m.e.p. is increased in most instances by as much as 50 per cent, or even more. Increased thermal loading due to turbocharging occurs only during operation at maximum power and can be alleviated by increasing the engine scavenge ratio. If the valve overlap period is increased to about 140 deg, both the inlet and exhaust valves are almost fully open at top dead centre, and the flow of scavenge air through the engine is considerably increased. A disadvantage of this arrangement in some engines is that part of the piston crown may have to be machined away locally to clear the valve heads: this may reduce the compression ratio and adversely affect the temperature distribution in the piston.

In Fig. 2, curves showing the air requirements of a typical engine are superimposed on the compressor delivery characteristics. From this illustration, it can be seen that turbocharger characteristics, in general, are such that they can be effectively matched to the engine requirements, provided the compressor has a high efficiency over the required range of delivery. The aim, therefore, is at matching so that the engine operates within the limits defined by the areas of high efficiency on the characteristic curves of the exhaust driven turbocharger that has been installed.

The B.S.A. Superchargers

A Range of Light, Compact Units, Suitable for Automotive and Other Diesel Engines

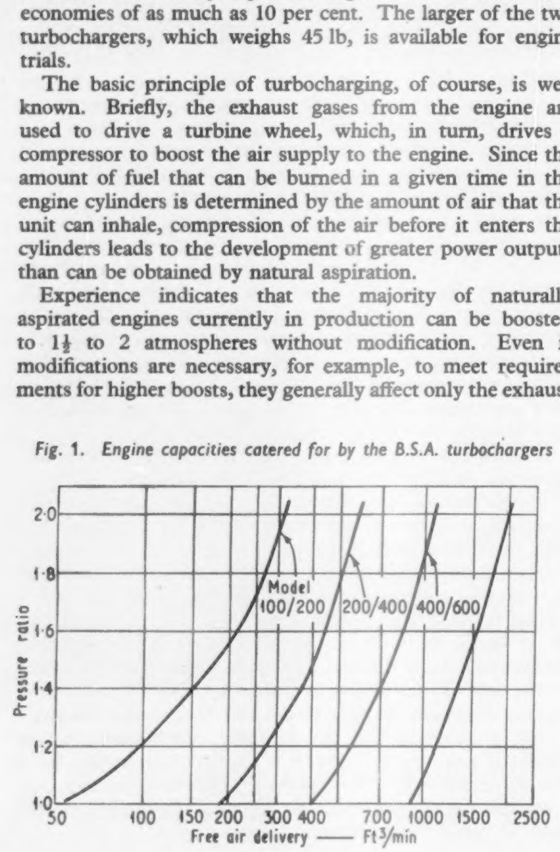
THE B.S.A. group of companies has been associated with the development of components for gas turbine engines since the earliest days of these power units. Much of their work in this connection has included the formulation of heat-resistant steels. In addition, they produce a wide variety of precision castings and components, such as turbine and compressor blades and discs. By virtue of this background of experience they have been well placed to develop their range of efficient and compact turbochargers. Moreover, by applying the techniques arising from modern developments in the group's manufacturing processes, they have been able to produce these turbochargers at a remarkably low cost.

So far, two sizes of turbocharger are ready for production. One is for diesel engines capable of developing from 200 to 400 b.h.p; the other, now being produced in batch quantities, is a smaller unit suitable for engines up to 200 b.h.p. This latter unit weighs 30 lb and its overall dimensions are approximately 8½ in diameter × 8 in long. Its maximum continuous rating is at a 2 : 1 pressure ratio and an exhaust temperature of 650 deg C. Trials have shown that an 8·6 litre engine equipped with this unit can be used to replace a 10·6 litre naturally aspirated engine with resultant fuel economies of as much as 10 per cent. The larger of the two turbochargers, which weighs 45 lb, is available for engine trials.

The basic principle of turbocharging, of course, is well known. Briefly, the exhaust gases from the engine are used to drive a turbine wheel, which, in turn, drives a compressor to boost the air supply to the engine. Since the amount of fuel that can be burned in a given time in the engine cylinders is determined by the amount of air that the unit can inhale, compression of the air before it enters the cylinders leads to the development of greater power outputs than can be obtained by natural aspiration.

Experience indicates that the majority of naturally aspirated engines currently in production can be boosted to 1½ to 2 atmospheres without modification. Even if modifications are necessary, for example, to meet requirements for higher boosts, they generally affect only the exhaust valves and piston assemblies, since bearing loadings are not greatly increased by supercharging. In any case, with very high-speed engines, the design criterion for bearings may be inertia rather than gas loading. There is no reason why engines designed specifically for turbocharging should not operate at boosts appreciably higher than 2 atmospheres.

Fig. 1. Engine capacities catered for by the B.S.A. turbochargers



Principal advantages

Since turbochargers generally are relatively small and light, the installed weight of turbocharged engines is far less than that of the naturally aspirated units giving the same power output. Similarly, an appreciable saving in installation space is possible, and the strength and weight of the frame required to carry the engine is also reduced. Moreover, an engine fitted with a turbocharger of economical design costs appreciably less than a naturally aspirated unit of the same power output.

Because of the improved overall efficiency of a turbocharged engine, the specific fuel consumption is lower than that of a naturally aspirated unit of the same power output. Two factors lead to this improvement in efficiency. One is that the mechanical losses in the turbocharged engine are not increased proportionately with the increase in power obtained by boosting the naturally aspirated version, while the second is that exhaust energy that otherwise would be wasted is used to boost the air flow to the cylinders, and this reduces the pumping losses of the engine.

The characteristic of the turbocharger can be matched to that of the engine to give the torque curve most suitable for the service for which the engine is to be employed. These requirements may be on the one hand for engines running continuously at maximum power output, or on the other hand for those requiring the highest torque output at low engine speeds. Only a very efficient turbocharger will meet the latter requirement satisfactorily.

Since, in operation, the speed of the turbocharger depends on the velocity of the exhaust gases passing through the turbine, the degree of turbocharging effected tends normally to increase automatically with engine speed and load. The overall noise level of a turbocharged engine generally can be reduced to a lower level than that of the naturally aspirated equivalent, because the exhaust gases, although entering the turbine as a series of high energy impulses, leave at a uniform velocity and a much reduced energy level. Similarly, the compressor smooths out the impulses caused by the opening and closing of the valves of the individual cylinders of the engine.

In general, the advantages of the turbocharged engine can be summarized as follows. Power output is increased in proportion to the degree of turbocharging. The overall power : weight ratio is considerably improved, and the installation space requirement for an engine of given output is reduced. Specific fuel consumption is reduced over a wide part of the operating range, and the initial cost of the engine per unit of power output is less than that of a naturally aspirated engine. Also, better torque-speed characteristics are obtainable. Another advantage, which is important for certain overseas markets, is that the power output obtained at sea level can be maintained at high altitudes. With the

B.S.A. turbocharger, because of the light weight of its rotating parts, there is an almost immediate response of the engine to changes of speed and load.

Economics of turbocharging

For a given output, the size of an engine varies inversely as the charge density; for instance, a six-cylinder engine turbocharged at a pressure ratio of 1·6 : 1 could, in certain applications, replace a naturally aspirated, eight-cylinder engine of the same bore and stroke. An even greater economy in size could be effected if the air charge, in passing between the turbocharger and the engine, were cooled to normal atmospheric temperature. Since the engine does not have to be altered structurally when a turbocharger is installed, its weight and size, and cost are not increased, so the additional cost is solely that of the turbocharger and its installation and the modified manifolds.

Quantitatively, the gains obtainable are indicated by the following data. A certain 200 b.h.p, naturally aspirated, high-speed diesel engine can be made to produce 270 b.h.p. simply by incorporating a turbocharger unit weighing only 30 lb. In the naturally aspirated condition, its specific weight is 12 lb/b.h.p, whereas in the turbocharged condition its specific weight is only 9 lb/b.h.p. So far as the engine alone is concerned, if its specific cost is £5 per b.h.p. in the naturally aspirated condition, its specific cost when turbocharged will be reduced to approximately £4 per b.h.p. As has already been mentioned, the economies obtained with regard to the frame and installation must also be taken into account when choosing between a turbocharged and a naturally aspirated engine.

The degree of fuel economy attained depends on the type of service for which the engine is used. Owing to its inherent flexibility, the B.S.A. turbocharger can be matched to give the best fuel consumption under the conditions in which the engine is mainly used. At speeds other than idling, this generally gives an improvement of between 5 and 10 per cent in specific consumption; between these limits, the figure actually obtained depends on the relative weight given to factors other than fuel economy, when the matching is carried out. Turbocharging should not have any adverse effects on maintenance requirements; in fact, for the larger size engines, the opposite has been claimed to have been experienced.

Matching the turbocharger to a diesel engine

The range of air flows obtainable with B.S.A. turbocharger units is shown in Fig. 1. From this, it can be seen that there should be no difficulty in selecting a turbocharger for any engine, provided its requirements are within the range of 50 to 2,000 ft³/min. It will probably be necessary, of course, to modify the exhaust and manifold porting to suit the turbocharger. The fuel injection equipment will also have to be adjusted to give the increased output, but it is important to avoid unduly lengthening the injection period. Apart from these, no other alterations should be necessary.

At first sight, it might appear to be surprising that the majority of engines currently in production can be supercharged to 1½ to 2 atmospheres, without major modifications. However, there are several reasons for this. One is that the maximum cylinder pressure can be controlled, within limits, by changes in injection timing or compression ratio. Secondly, the higher temperature and density of the charge at the end of the compression stroke increase the rate of heat transfer to the fuel droplets injected into the cylinders: consequently, the ignition delay and the high rate of pressure rise associated with it are reduced. This tends to reduce the shock loading on the bearings and is also one of the factors that helps to make turbocharged engines quieter than naturally aspirated units. A further advantage associated

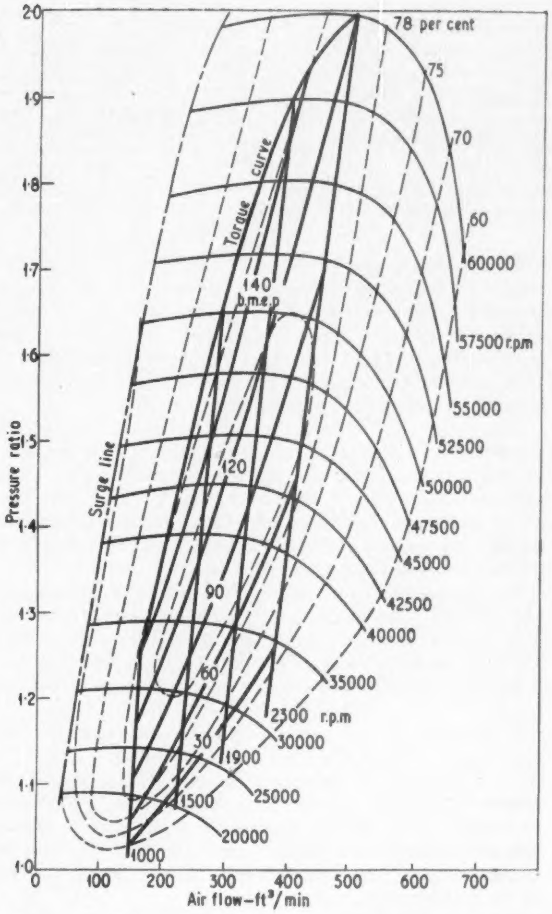


Fig. 2. The matching of the engine and compressor air flow characteristics, which are shown in heavy and light lines respectively

with this improvement in combustion characteristics is that lower quality fuels can be employed.

Turbocharging generally increases the maximum cylinder pressure about 200 lb/in² above that of naturally aspirated engines, which is commonly in the order of 1,100 lb/in². Despite this relatively small increase in maximum pressure, the b.m.e.p. is increased in most instances by as much as 50 per cent, or even more. Increased thermal loading due to turbocharging occurs only during operation at maximum power and can be alleviated by increasing the engine scavenge ratio. If the valve overlap period is increased to about 140 deg, both the inlet and exhaust valves are almost fully open at top dead centre, and the flow of scavenge air through the engine is considerably increased. A disadvantage of this arrangement in some engines is that part of the piston crown may have to be machined away locally to clear the valve heads: this may reduce the compression ratio and adversely affect the temperature distribution in the piston.

In Fig. 2, curves showing the air requirements of a typical engine are superimposed on the compressor delivery characteristics. From this illustration, it can be seen that turbocharger characteristics, in general, are such that they can be effectively matched to the engine requirements, provided the compressor has a high efficiency over the required range of delivery. The aim, therefore, is at matching so that the engine operates within the limits defined by the areas of high efficiency on the characteristic curves of the exhaust driven turbocharger that has been installed.

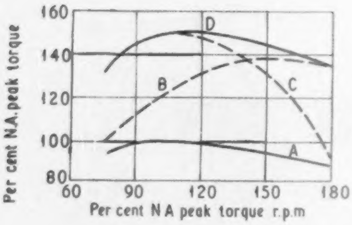


Fig. 3. Typical curves

A *naturally aspirated;* **B** *turbocharger matched at maximum engine speed;* **C** *matched for the temperature at the lowest speed and governed to give at maximum speed the same power output as if naturally aspirated;* **D** *matched for temperature at the lowest speed and arranged to maintain this turbocharger speed up to the maximum engine speed*

This matching is carried out in two ways. First, adjustments are made, by controlling the velocity of the exhaust gas flow through the turbine, to regulate the rotational speed of the turbocharger; this is done simply by selection of the most suitable turbine nozzle ring. During the initial matching tests, this operation can be carried out on the test installation without removing the turbocharger.

The second adjustment that can be made concerns the compressor. A range of rotors and diffusers is available, and any of this range can be accommodated simply by fitting different distance rings between the flanges of the two parts of the collector, or volute. When the air requirements of the engine are known accurately, it may even be possible to dispense with advance matching tests.

For automotive applications, a turbocharger operating at high efficiency at relatively low engine speeds is desirable. Matching to meet this requirement causes the engine to have a high excess air factor when operating at high power outputs. Therefore, turbocharged engines tend to operate at constant b.h.p.

The shape of the torque curve is determined by the matching of the engine and the turbocharger. Normally, the turbocharger is matched at a given engine speed and exhaust temperature. At lower speeds, exhaust temperature or smoke, limits the permissible boost pressure or turbocharger speed, while at higher speeds the limit is set by the maximum safe operational speed of the rotating components of the turbocharger.

The shape of the torque curve required depends, of course, on the service conditions. For an industrial or marine engine, a curve that rises with engine speed is called for, so the engine and turbocharger should be matched at maximum engine r.p.m; this is not a difficult condition to satisfy. The requirements of road transport vehicles, on the other hand, vary according to the type of operation, although, in general, a curve that rises as engine speed falls is desirable. City transport vehicles, which are continually stopping and starting, require maximum torque at lower engine r.p.m. than long-distance, high-speed passenger vehicles. It follows that the governor control on road vehicles is more complicated than that required for marine and industrial applications. The torque curves shown in Fig. 3 illustrate different conditions of matching with a 2 : 1 ratio turbocharger. Fuel economies effected are, in general, proportional to the increase in b.m.e.p. obtained by turbocharging.

Two-stroke engines have advantages where a high degree of supercharging is required. This is because the cylinder size is smaller, for a given power output, than that of the four-stroke type of engine, and this gives better rigidity and heat flow conditions. In addition, the cool air passed through the cylinders for scavenging purposes tends to reduce thermal loading.

Low-speed, two-stroke engines with well designed porting have been operated with an efficient turbocharger without an ancillary scavenging blower. However, some means of augmenting the turbocharger at starting and light load conditions may be necessary, particularly for the small high-speed units. To this end, engines have been operated with a positive displacement compressor in addition to a turbocharger.

With this arrangement, the compressor and the turbocharger may be installed in series. A relatively small, positive displacement blower can be employed; it becomes unloaded as the engine output is increased and the turbocharger takes over. In fact, at the higher outputs, the crankshaft-driven blower may act as an expander and therefore cool the engine air supply and return power to the crankshaft. An alternative arrangement is to install the positive displacement blower in parallel with the turbocharger. With this layout, a clutch may be incorporated in the drive to the positive displacement blower, which can thus be disengaged when the turbocharger takes over.

Layout of the unit

The layout of the B.S.A. turbocharger can be seen from Fig. 4. Two housings, one for the compressor and the other for the turbine, are assembled on each end of the central aluminium die casting. All three components are held together by eleven bolts that are passed through the unit from end to end. The rigid, cast iron housing for the bearings that carry the rotating assembly is mounted on studs on the central aluminium die casting. Thus, it is a relatively simple matter to remove the compressor and turbine housings, for inspection and cleaning during service, without disturbing the rotating assembly. Both the compressor rotor and its two-piece housing are aluminium die castings. A two-piece casting of chromium-silicon iron houses the turbine rotor, which is of Jessops G18B steel. The rotor and its housing can be operated, without risk of failure, at temperatures of up to 750 deg C. Clamped between the two pieces of the housing is the nozzle ring, which is of heat-resistant steel.

An important requirement for a unit intended for automotive application is, of course, that the weight of its rotating components should be as low as possible. In the unit illustrated, which has a 4 in diameter compressor rotor and a $3\frac{13}{16}$ in diameter turbine rotor, the weight of the rotating components is only $2\frac{1}{2}$ lb, and its polar moment of inertia is 7·1 lb-in^2. Both rotors are connected to their tubular shaft by means of Hirth couplings. The whole assembly is held together by an En24 stud screwed into the turbine rotor, the nut being tightened against the boss of the compressor rotor.

Two, single-row ball bearings, with their centres spaced about $1\frac{7}{8}$ in apart, carry the rotor assembly. Their inner races are separated by a tubular distance piece and their outer races are 0·002 in clearance fit in their housings. This clearance is taken up by a ring of silicone-rubber—a material chosen because of its resistance to deterioration at relatively high temperatures—in an annular groove in the housing. Thus, the bearings are free to move 0·001 in radially from their truly central position in their housings, the movement being damped by the silicone-rubber rings. This arrangement has been adopted to prevent vibration of the bearings, which tends to occur at about 36,000 r.p.m. Location is effected at the bearing adjacent to the compressor. The outer race of this bearing is clamped between the compressor casing and a snap ring in a groove in its housing. A light aluminium shroud encloses the bearing housing and its oil feed system.

Both bearings are lubricated and cooled by oil squirted into them from a nozzle inserted between them from the top of the casing. Provision is made at the upper end of the nozzle to prevent particles of foreign matter from passing into and blocking the $\frac{3}{64}$ in diameter jets drilled in its lower end. The oil flows through the jets at the rate of 20 gal/hr, and it drains away through a large diameter connection in the base of the housing. It has been found that the maximum temperature of the inner race of the bearing adjacent to the turbine is 150 deg C. In some of the early experimental models, the bearings were lubricated from an oil sump incorporated in the turbocharger, but this was considered to

be unsatisfactory because the capacity of the sump was not sufficient to give an adequate margin of safety under normal operating conditions.

Leakage of gas from the turbine and compressor rotors along the shaft is minimized by the employment of labyrinth seals, while oil thrower rings prevent lubricant from splashing on to the seals. Restriction of heat flow between the turbine and the compressor has been effected in two ways. First, a polished aluminium radiation shield is interposed between the turbine and the central portion of the casing. Secondly, the area of contact between the turbine casing and the remainder of the unit is minimized, insulation ducts being cored in the face to which the flange of the turbine casing is bolted. Tests have shown that heat flow from the turbine to the compressor increases the temperature of the air passing through the compressor by no more than 5 deg C.

Installation

When the unit is installed close to the engine and cast iron manifolds are employed, the turbocharger can be mounted directly on the manifold flange. If the unit has to be mounted further away from the engine, it can be carried on a bracket; for this purpose, rigid mounting points are provided at each side of the central portion of the casing. Bends and other features liable to cause losses in the ducting should be avoided so far as possible. If practicable, the unit should be mounted with its axis horizontal and its pitch and roll angles should not exceed 30 deg for oil drainage.

Turbine casings either with one or two entries are available, and it is essential that the connections to these casings should be so arranged as to offer a minimum of interference with the exhaust flow. The angular relationship between the compressor and turbine casings is unimportant so far as the operation of the turbocharger is concerned; they are designed so that they can be fitted together at different relative positions 30 deg apart round the axis of the shaft.

The oil feed pipe that serves the bearings of the turbocharger is generally taken from the engine filter. This pipe should have a bore of $\frac{5}{16}$ in diameter. A drain pipe of at least $\frac{7}{16}$ in bore is also required; its connection to the engine should be near the crankcase breather.

Since the turbocharger increases the air flow through the engine, a larger air cleaner is required for the turbocharged engine than for the naturally aspirated version. The size required varies according to the degree of supercharging. If the air cleaner is required to be part of the turbocharger, suitable mounting arrangements can be incorporated on the compressor and central casings. Otherwise a hose connection is made between the compressor inlet and the duct from the air cleaner. A similar type of connection is used at the compressor outlet. Back pressure in the exhaust system should be kept to a minimum. In this connection, it should be borne in mind that, by virtue of the use of a turbocharger, a simplified exhaust silencer can generally be used.

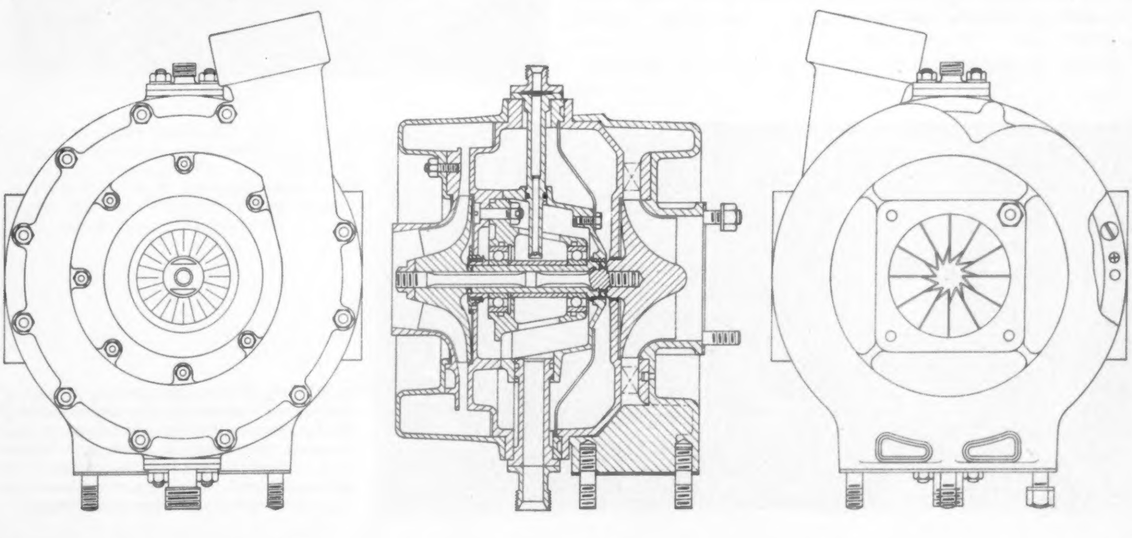
Experience has shown that the unit can be run for periods of at least 1,000 hr without being cleaned. When cleaning is required, it is necessary only to disconnect the pipes to the compressor and turbine, and remove the casings. This exposes the turbine and compressor rotors. In replacing the casings, care should be taken to avoid damaging the rotors. A check should be made to ensure that the rotors run freely after the casings have been replaced.

Design features

Early tests on small, radial flow compressors and turbines showed their efficiencies to be higher than expected as a result of theoretical analyses. Development has now reached the stage where efficiencies equal to those obtained from much larger units are the general rule. The principal advantage of the radial flow type of turbine is that the rotor can be produced relatively cheaply by casting, whereas the axial flow type is much more difficult to manufacture. For most small engines, the gas ducting problem is simpler in the radial flow type of compressor and turbine installation than in an axial flow type. Also, the simplicity of form and high strength : weight ratio of the radial flow turbine are marked advantages. In fact, the advantages of the radial flow units are so pronounced that for an engine requiring a greater air flow than can be provided by a small turbocharger, it would be more economical to install two units of this type rather than one axial flow type.

In larger sizes, the rule is almost invariably to use a centrifugal compressor in conjunction with an axial flow turbine. It would appear that the dimensional limit below which radial flow turbines become more efficient than the axial flow type is a blade tip diameter of about $7\frac{1}{2}$ in and a blade length of approximately $\frac{3}{4}$ in. This means that, in terms of engine size, units of 600-700 b.h.p. and more are

Fig. 4. Noteworthy features of the B.S.A. turbocharger are its compactness, the light weight of its rotating assemblies, and its simplicity of design



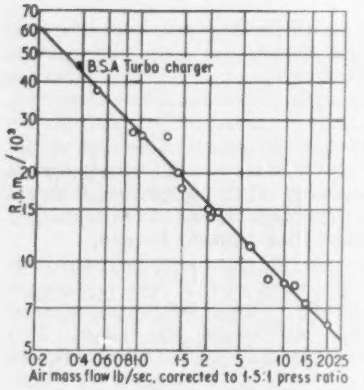


Fig. 5. Curve showing rotational speed plotted against mass flow of air for a variety of compressors known to operate satisfactorily

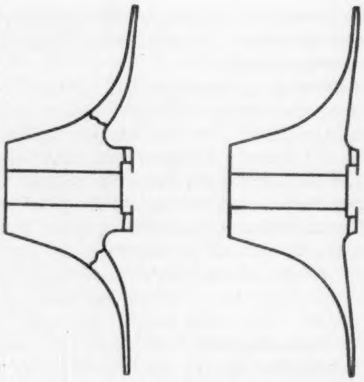


Fig. 6. Left: Typical failure obtained in bursting tests. Right: Design change to increase bursting speed by flattening the disc

probably better served by an axial flow turbine. For larger units, the axial flow turbine tends to be lighter than the radial flow type; this difference in weight becomes progressively less marked as size is reduced. Stresses due to temperature differentials are liable to be higher in the axial flow type, since the disc is not in contact with the main flow of exhaust gas; on the other hand, the mean temperature is lower than that of the radial type.

Early in the design stage of the B.S.A. units, many basic rotating assembly layouts were considered. For example, the two rotors could have been mounted back-to-back on their shafts and overhung from a pair of bearings. Advantages of this layout are that the axis of the shaft can be arranged vertically or horizontally, as required, and the bearings can be incorporated on the cold side of the unit.

A disadvantage is that the efficiency of the compressor may be reduced by heat transfer from the turbine. This can be avoided by diverting some of the air from the compressor for cooling purposes and by interposing heat insulation between the rotors. Another disadvantage is that with this arrangement it is difficult to make provision for cleaning the compressor without disturbing the rotating assembly. In addition, the bearing housings tend to obstruct the air passage to the compressor eye. Although the natural frequency of this system is lower than that of any other arrangement, this disadvantage can be overcome by ensuring that the frequency is below the normal range of operating speeds of the unit.

Another arrangement considered was that of the two rotors mounted between the bearings. For small radial flow turbines, this would be practicable only if the bearing at one end could be housed in such a way that it would not be adversely affected by the passage of hot exhaust gases past it.

In the layout that was actually adopted, the two bearings are interposed between the rotors. This not only has the advantage, already mentioned, that the blower and turbine casings can be removed for cleaning without disturbing the rotating assembly, but also the bearings in no way obstruct the turbine and compressor ducting. Its whirling speed is in excess of 80,000 r.p.m.

The critical features of the whole design are the bearings and the rotors, since these are the only potential source of failure. Out-of-balance loads would be liable to cause fatigue failures of the inner races of the bearings. While the balancing of the rotors on their shaft can be effected accurately on assembly, this would be of no avail if the parts did not remain in balance during service. The balance might be disturbed by temperature gradients, local yielding of the material or slight relative movement of the components. These problems have been overcome by the employment of Hirth couplings to connect the rotors to the shaft. With this arrangement, yielding can take place without distortion, while at the same time, the original location is maintained more accurately than would be practicable with spigot or

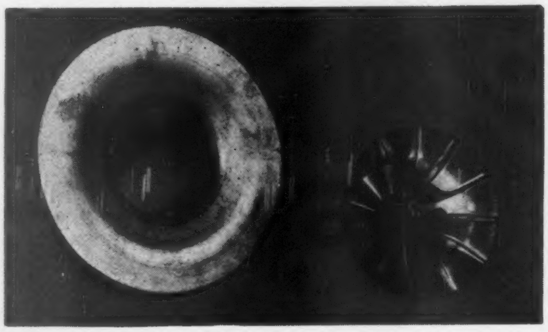


Fig. 7. Above: The blower rotor, shown here with the diffuser, is an aluminium, pressure die casting. The only machining on it, apart from that necessary to form the Hirth coupling, is the removal of the thin flashes that remain on the edges of the radial blades

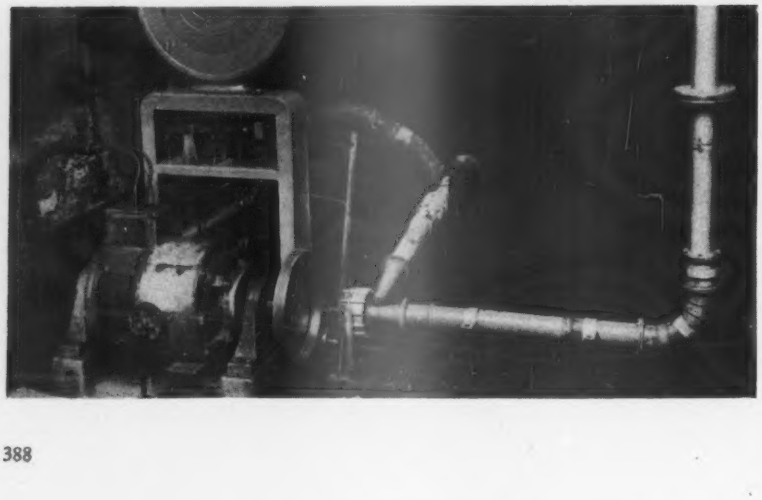


Fig. 8. Left: In this high-speed rig for testing compressors, a variable-speed motor, developing 50 b.h.p, with a 13:1 ratio belt-drive, is used to operate compressors at speeds up to 65,000 r.p.m. The belt-drive has been found to be satisfactory, and does not give trouble at high speeds as did the gearbox used formerly

face joints. Accurate control of the tension in the central bolt that holds the rotating components together is most important.

Although the natural frequency, or whirling speed, of the rotating assembly is estimated to be well above the running speed of the unit, the bearings themselves tend to vibrate. Static tests and calculations show that these vibrations can be attributed to deflections due to local pressures between the balls and their tracks. These vibrations are damped, as has already been mentioned, by mounting the outer races in silicone-rubber rings, and allowing them a limited amount of freedom.

There is no difficulty in designing the rotors in such a way that they will not rupture. This can be deduced from two facts: first, turbine rotors of the axial flow type have been operated satisfactorily at appreciably higher peripheral velocities than in this turbine, secondly, the radial flow type of rotor is fundamentally more robust in construction, not only because the blades are cast integrally with the disc, but also because of their radial arrangement. The B.S.A. turbine disc is particularly robust because it is not drilled through the centre and is made of the best heat-resistant materials available.

Since both the compressor and turbine blades are cast integrally with the rotors, their natural frequencies of vibration are extremely high and are therefore unlikely to be a source of trouble. In the compressor, the clearance between the rotor blades and the casing is 0·007-0·009 in and in the turbine it is 0·013-0·015 in. The weight of the whole rotating assembly, including the races of the inner ball bearings, is 2¼ lb.

DEVELOPMENT

Compressor

Much of the development story has already been told by C. A. Judson and E. Kellett in a paper presented at the Symposium on Superchargers and Supercharging, held early this year by the Automobile Division of the Institution of Mechanical Engineers. When the design was first contemplated, there was very little information available on small radial flow turbines. However, a great deal was known about centrifugal compressors, which, in general, present many more difficult design features than do turbines. For these reasons, the compressor was made the basis of the design, since its performance relative to speed could be predicted with reasonable accuracy. For a given compression ratio, the compressor speed is fixed by the rotor diameter; if the rotor is too large, the frictional losses become excessive, and if too small, the gas flow pattern is unsatisfactory.

From Fig. 5, in which rotor speed is plotted against mass flow for a number of different compressors known to be satisfactory, it can be seen that, for a compressor of the size

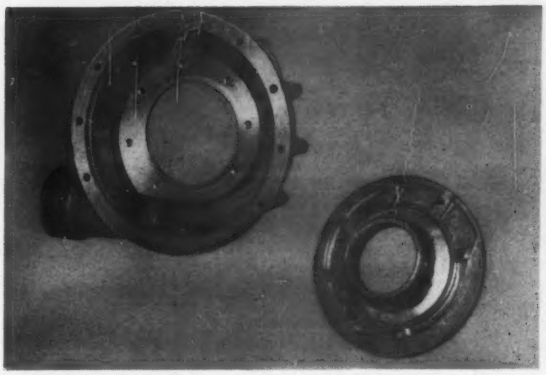


Fig. 9. The volute is of rectangular section, so that it can be pressure die cast. To conserve space, its outer periphery is of circular form, while the inner periphery follows a volute curve

of the B.S.A. unit, optimum performance should be obtained at rotational speeds of 40,000-50,000 r.p.m. However, it was at first thought that, to obtain long periods between overhauls, the shaft speed would have to be limited to about 32,000 r.p.m.; therefore, a rotor of 5 in diameter was employed. The unit designed on this basis operated satisfactorily, but was rather bulky and heavy. Consequently, a second design, with rotors of 4 in diameter, was built for operation at 45,000 r.p.m. This unit proved to be more satisfactory so far as performance was concerned and presented no problems with regard to service.

Early investigations showed that compressor efficiencies of about 80 per cent should be practicable, and a comprehensive programme of development was planned on this basis. The aim was at developing a unit with high efficiency over a wide range of air flow. First, investigations were carried out to determine the performance of rotors of different profiles. Tests were also carried out to determine the effects of varying the relative lengths of the radial and axial portions of the flow paths between the blades, and also of varying the thickness of the blades. The form finally adopted can be seen in Figs. 4 and 7.

A great deal of development was carried out on the form of volute of the collector, shown in Fig. 9. The aim was to maintain a constant and a minimum outside diameter and effect the area changes, to give a volute, by alteration to the radius of the inner wall and to the axial width of the space to the end wall. Ramps of various forms were used to assess the effects of relative changes to these dimensions and the final form was of constant outside diameter, constant axial depth and of a varying radius to the inside wall. Its advantages, from the installation point of view, of course, are

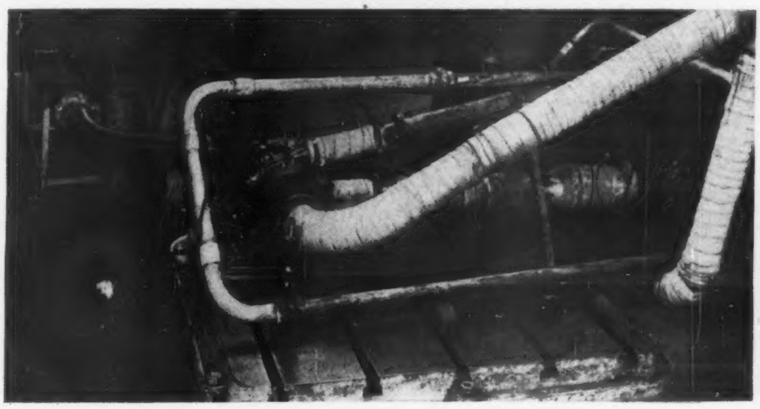


Fig. 10. On this self-running rig the compressor outlet is connected to an aircraft gas turbine combustion chamber and the outlet from the chamber is connected to the turbine inlet. The horizontal pipe joining that between the compressor outlet and the combustion chamber is from an external air supply for starting

Fig. 11. On the left is the master pattern, used on a pantograph milling machine, in the production of the semi-circular brass insert in the component shown in the centre. This component, together with its lid, on the right, forms a mould into which the stone, in front, is poured. The cavity in the semi-circular brass insert is the shape of the space between each pair of rotor blades, thus the stone is in effect a core

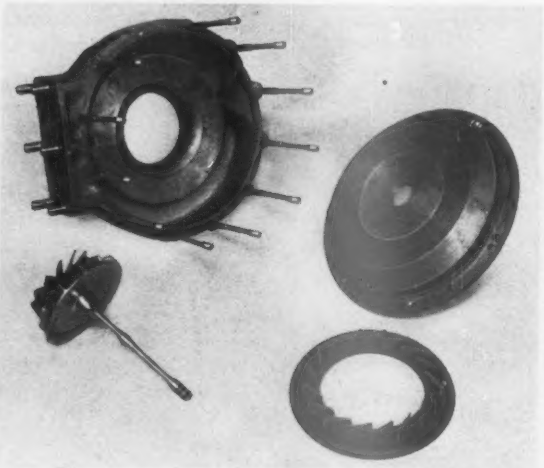
obvious, and it also enables the compressor casing to be pressure die cast. Bladed diffusers were tested, but free vortex diffusion was found to give similar efficiencies over a wide range of mass flow. The blower test rig is illustrated in Fig. 8.

Over-speed tests showed that the bursting speed of the compressor rotor is probably higher than 100,000 r.p.m. To obtain failures in these tests, rotors were made in a low grade material. From the failures thus obtained, it was possible to determine design changes desirable to improve the strength of the unit. The modifications subsequently made increased the fracture speeds by about 10 per cent. They involved designing the rotor disc in such a way that its end face remote from the blades was of very obtuse-angle conical form, rather than bell-shaped, as can be seen in Fig. 6.

Turbine

Development testing of the turbine was not begun until the compressor characteristics were firmly established. This was in order that the compressor could be used as a brake. For these tests, a self-running rig incorporating an aircraft gas turbine combustion chamber was employed. The arrangement of this rig is shown in Fig. 10. Air goes through

Fig. 13. One stud is omitted from the turbine casing to accommodate the blower outlet on the other side of the unit. The angular relationship between the blower and compressor casings can be adjusted in intervals of 30 deg to suit the installation. On this experimental nozzle ring, the blades are dowelled on, but production rings are cast in one piece by the wax investment, sand casting method of manufacture



a larger silencer to the compressor inlet; thence, it is ducted from the compressor outlet to the combustion chamber. A pipe, from the starting air supply, is also connected to the duct between the blower outlet and the combustion chamber. Finally, the outlet from the combustion chamber is ducted to the turbine inlet, and the turbine outlet is connected to an exhaust pipe leading out of the shop. At first, combustion was started by using a mains air supply; however, it was found that the drain on this supply was inconveniently large, so a special rig, comprising an aircraft supercharger driven by a 200 b.h.p. engine, was installed.

After a number of experimental rotors, machined from billets and forgings, had been produced, a technique was evolved for casting rotors economically. This removed limitations on the design of the rotor passages. The fact that on each rotor the vanes and passages between them are identical enables a relatively simple technique to be employed. It comprises the preparation of a number of identical cores, or *stones*, representing the air flow passages between the blades, and to assemble them radially in a circular box in such a way that the complete assembly forms a mould for the preparation of a wax pattern.

In detail, the procedure is as follows. First, a steel pattern is made to the exact shape of the space between adjacent

Fig. 12. This illustration shows some of the components of a mould in which wax patterns for a rotor are cast. The stones are each cast on to a small steel base, by means of which they are secured on the circular frame, to which they are individually dowel-located

blades of the rotor, but to a larger scale. This pattern, as can be seen from Fig. 11, comprises three parts: one represents the back plate of the rotor, and is turned in a lathe, and the other two are cheeks to represent the blade profiles. For blades that are truly radial in form, these cheeks are machined on a shaper, but if their profiles are of double-curvature, they are built up from steel laminae, the edges of which are machined to the appropriate contour. The steps between the edges of each of the laminae are filled with a plastic metal, called Devcon, to form the continuous profile required.

This pattern is then reproduced to the correct scale, in brass, by means of a pantograph milling machine. The milled pattern then forms part of a mould. There are four other parts: two cheeks, a base plate and a lid, assembled as shown in Fig. 12. A material called Titanite is poured into this mould. Titanite is a powder, which is mixed with water immediately before use. After it has been poured into the mould, it sets hard and resembles stone. Hence, the term stone is given to the components made with it. The finished stone is robust and will stand a reasonable degree of rough handling. It also has the advantage that a good surface finish is obtainable.

Each stone is cast and wired on a separate steel base, which, before the Titanite is poured, is located relative to the pattern by a dowel. These bases, with the stones on them, are assembled radially on to a circular steel frame and retained by screws. The complete assembly is then placed in a box, the bottom of which is shaped to form the end face of the

back plate of the rotor. There is a riser in the top, or lid, through which the wax is poured.

The turbine rotor failures that were experienced occurred at the blade roots and were initiated by cracks at the run-out of the trailing edge of the blades. These defects were cured by changing from a simple fillet radius to a hyperbolic section at the junctions between the turbine disc and the blades. Failures were also caused by inaccurate spacing of the cores, giving inconsistent blade thickness. Since creep had to be taken into account, exhaustive hot tests were required to determine the safe speed of the turbine. These tests indicated that, from the purely mechanical point of view, the speed of the unit would be limited primarily by the turbine.

While a great deal of information was obtained from rig tests, final development had to be carried out on engines, because it was not practicable in any other way to reproduce the pressure pulses experienced in the engine installation. Casings with one, two or three entries were tried, to suit different engine layouts. In all instances, the ducting to the turbine was arranged so that interference between the pulses was minimized. For example, the exhaust manifold of a six-cylinder engine is generally required to be divided in such a way as to serve two groups of three cylinders, Fig. 17.

Two types of turbine casing were finally evolved. One is a twin tangential-entry casing, each entry being of true volute form. The second is a single-entry, divided-flow casing with no volute, Fig. 13. For development, baffles were fitted in the casings to determine the most suitable aerodynamic form.

Grey-iron nozzle rings were used for experimental purposes, but for normal operation over long periods, a medium grade, heat-resistant steel, with a good scale resistance is required. Tests with turbine nozzles of various forms showed that blade section had little effect on performance, except inasmuch as there was a slight improvement as the thickness of the blades was reduced.

The procedure for the evaluation of nozzle ring performance was as follows. Different rings were fitted in turn to a turbocharger installed on a specific engine, and the compressor pressure ratios obtained with each were plotted against b.m.e.p. at constant engine speed and matching conditions. Turbine blade exit angles were adjusted to reduce to a minimum the swirl in the outlet. It was found that swirl could be reduced by increasing the length of the curved path between the vanes.

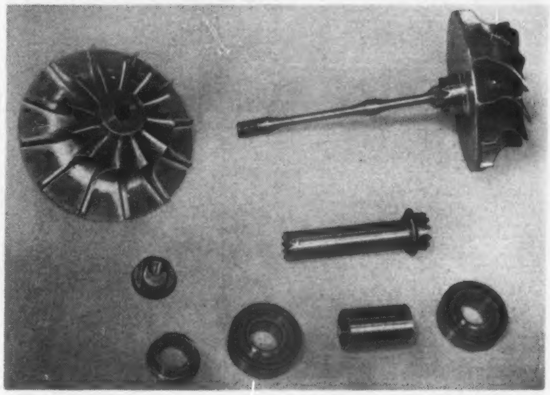


Fig. 14. For experimental purposes, there are two flats on the nut that is tightened against the compressor rotor to hold the rotating assembly together. One is to reflect a light ray on to an electronic counter, while the other is simply to maintain the balance. The whole of the nut, except the reflecting flat, is painted black. The shaft is ground to a tolerance of 0·0002 in to avoid straining the inner races of the supercharger grade, that is, specially selected ball bearings

Rotating assembly

Because of the high rotational speeds of this unit, a number of new design problems were introduced. Location of the rotor relative to the shaft was critical, since a 1 lb rotor eccentrically mounted only 0·001 in would produce at 45,000 r.p.m. an out-of-balance force of almost 60 lb. Spigot location was not considered to be satisfactory, because the fit of the spigot is liable to alter as the stresses in the rotor increase. It was also considered undesirable to have a hole in the centre of the turbine for location purposes, since the turbine stresses were expected to limit the performance of the unit. The Hirth coupling arrangement finally adopted has the advantage that location is accurately maintained, even if the rotor stretches appreciably.

Among the bearings tested were sleeve type, spring-loaded

Fig. 15. The aluminium alloy centre casing illustrated was manufactured for experimental purposes and therefore is a sand casting, but it has been designed for pressure die casting. A rigid iron casting, centre, forms the housing for the bearing assembly; it is enclosed by the aluminium alloy cover, shown to the right of the centre casing

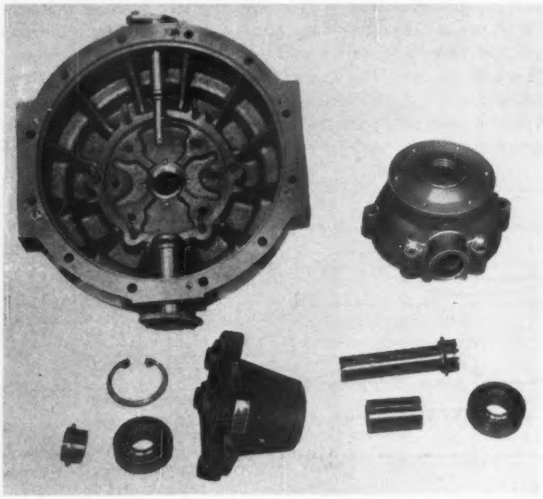


Fig. 16. The high-speed test rig used for development work on bearings. A 5 b.h.p. electric motor is employed, and it operates at speeds up to 75,000 r.p.m. The coupling between the motor and the component under test is of nylon, which does not fret seriously, and has the flexibility and strength necessary for this very high speed application

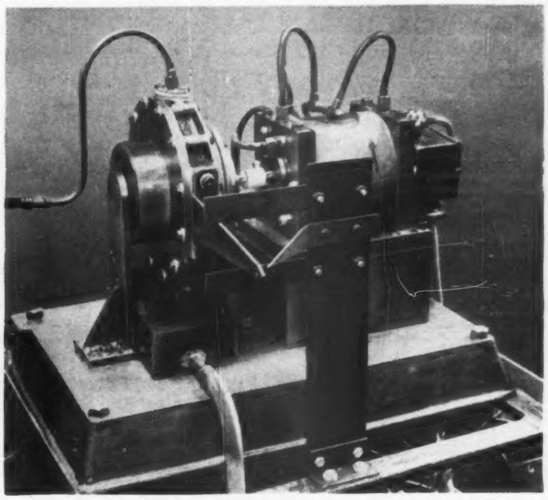


Fig. 17. A test bench installation of a double-entry turbocharger, showing the arrangement of the exhaust manifolds

angular contact, and standard ball bearings. Plain bearings were the least successful, and tests at 60,000 r.p.m. led to failures due to fretting and even peeling of the white metal from its steel backing. Frictional losses were higher with this type than with standard ball bearings. It is of interest to note that bronze bearings, impregnated with polytetrafluoroethylene, PTFE, have been run successfully for short periods in the unlubricated condition at 75,000 r.p.m. in water-cooled housings.

When angular contact bearings are used, they have to be preloaded. In the turbocharger, since the axial thrusts of the compressor and turbine rotors oppose each other, this preload can be fairly low. However, when testing the compressor alone, it was necessary to apply a preload of 60 lb. Under these conditions, frictional heating caused many failures of the inner race at speeds of approximately 75,000 r.p.m. This failure could be avoided either by reducing the contact pressure or by supplying extra oil to cool the bearing shaft.

The only trouble experienced with standard ball bearings at these speeds concerned the cages. Subsequently, the bearing manufacturers supplied bearings with more robust cages; this eliminated the trouble. As a result of this development work, standard ball bearings, specially selected for accuracy, are now fitted. If it should be decided later to manufacture turbines to operate at even higher speeds, it is possible that angular contact bearings may have to be used.

The rotating assembly is illustrated in Fig. 14, while the bearings and their housing, together with the centre casing, are shown in Fig. 15. Much of the development work on the bearings and their lubrication system was done with the high speed rig, Fig. 16. Mounted on this rig is a centre casing assembly, with a torque reaction arm bolted to it. A nylon sleeve couples the bearing shaft to the high speed electric motor. This material was chosen because of its flexibility, strength and the freedom from fretting trouble experienced with it; couplings of various metallic materials were tried previously, but proved to be unsatisfactory at very high speeds.

Testing in vehicles

A series of tests has been carried out with standard double-deck buses operating on normal service routes, but with a constant load of just over 10 tons. The vehicles used for these tests were virtually identical, except for the engines. A Daimler CD650, 10·6 litre engine was employed for the naturally aspirated installation and a Daimler CVD6, 8·6 litre engine was used with the turbocharger. The characteristics of the turbo-charged 8·6 litre engine are shown in Figs. 20 and 24. In the Fig. 20, the bus rating, which, of course, is not at the full rack position, is shown by the chain-dotted curve.

In all tests, the two vehicles were run over identical routes and operated normally; in fact, one preceded the other by about a quarter of an hour. The drivers changed places half-way through each test, the vehicles running twice over each test route. At all regular stops, the buses were kept stationary with their engines idling for 10 seconds. This was done to maintain a degree of control over the operating conditions to enable reasonably accurate comparisons to be made.

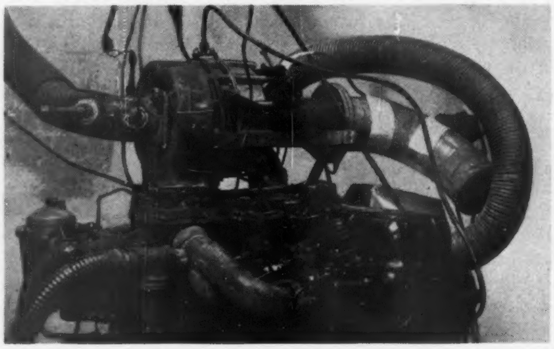


Fig. 18. A turbocharger with a divided, single-entry turbine, installed with an intercooler on a 9·6 litre engine. The pipes connected to the exhaust manifolds are to bleed off exhaust gas, as necessary, to prevent running over the 2:1 pressure ratio at high engine speeds with a turbocharger matched for low speeds

The tests were carried out on four routes. On route 1, the steepest gradient was 1 in 7, the highest point 850 ft, the lowest point 275 ft above sea level, and the total length of the test run was 20·98 miles. The steepest gradient on route 2 was 1 in 18, and the highest and lowest points 760 ft and 300 ft respectively; the length of this route was 33·9 miles. Route 3 covered a total distance of 16·82 miles, in which the steepest gradient was 1 in 10, the highest point was 1,150 ft

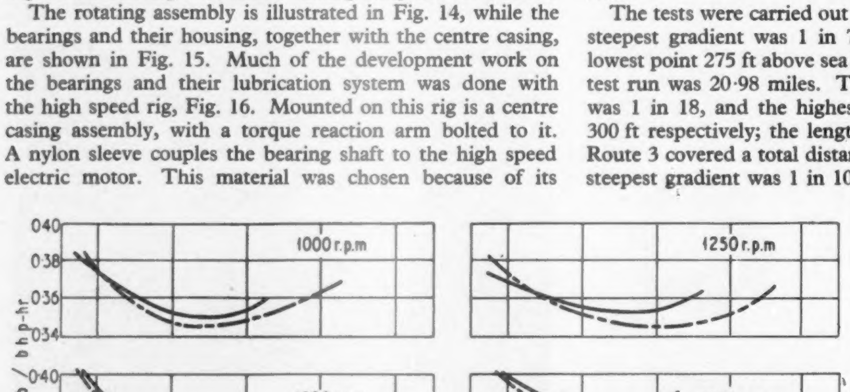
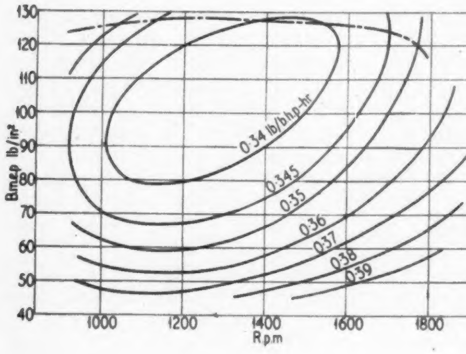
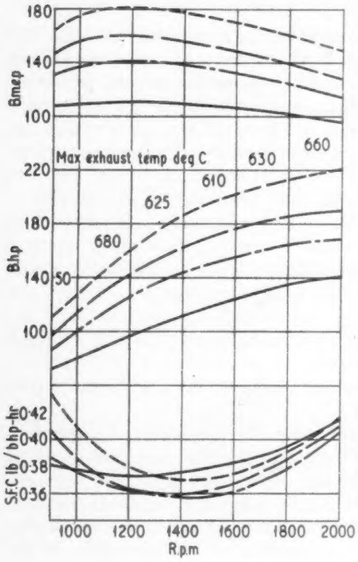


Fig. 19. Fuel consumption loops for an engine in the naturally aspirated and turbocharged conditions



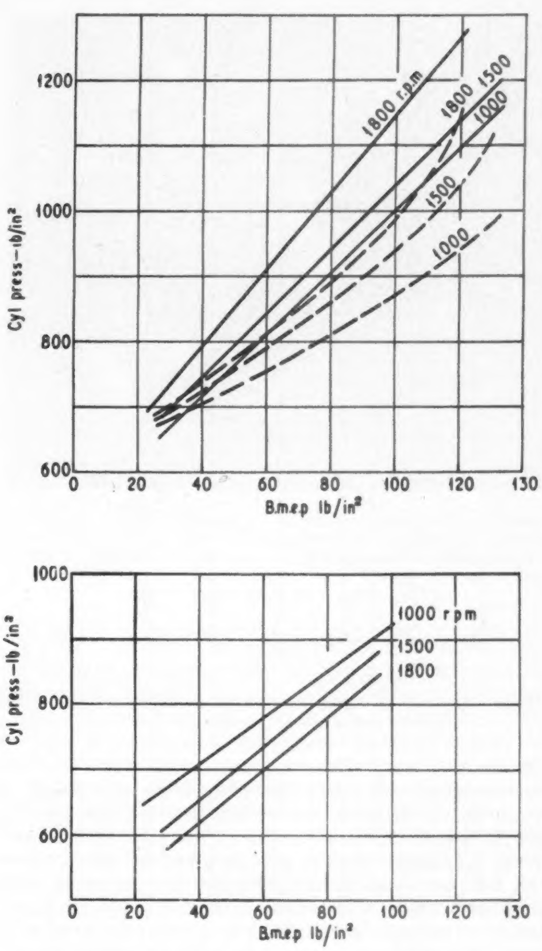


— — — — — — Bus rating

Fig. 20. Above: Characteristics of an 8·6 litre turbocharged engine

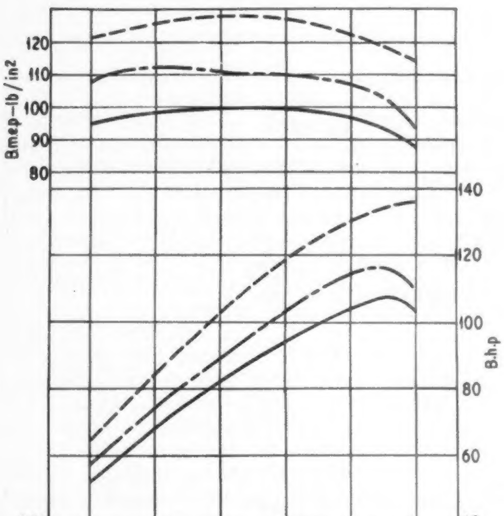
——— Naturally aspirated
– – – – / — — — / —— —— Turbocharged with fuel pump rack set at three different fixed positions

Fig. 21. Left: Performance curves for a 9·6 litre engine turbocharged to 2 atmospheres and equipped with an intercooler installed



——— Standard timing, 27 deg B.T.D.C. – – – Timing retarded 6 deg

Fig. 22. Peak cylinder pressures obtained with the Daimler CD6, 8·6 litre engine: Above, turbocharged to 2 atmospheres, and below, in the naturally aspirated condition

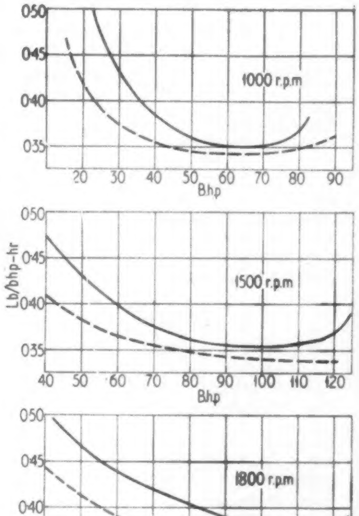


——— Naturally aspirated
— — — 1·5:1 pressure ratio
– – – – 2:1 pressure ratio

Fig. 23. Left: Comparison of performance curves for the Daimler CD6 engine in the naturally aspirated condition and turbocharged for maximum fuel economy and smoke-free operation throughout the entire range

——— 10·6 litre engine, naturally aspirated
– – – – 8·6 litre engine, turbocharged

Fig. 24. Right: Comparison of the brake specific fuel consumptions of a 10·6 litre naturally aspirated engine and an 8·6 litre unit turbocharged to perform the same duty



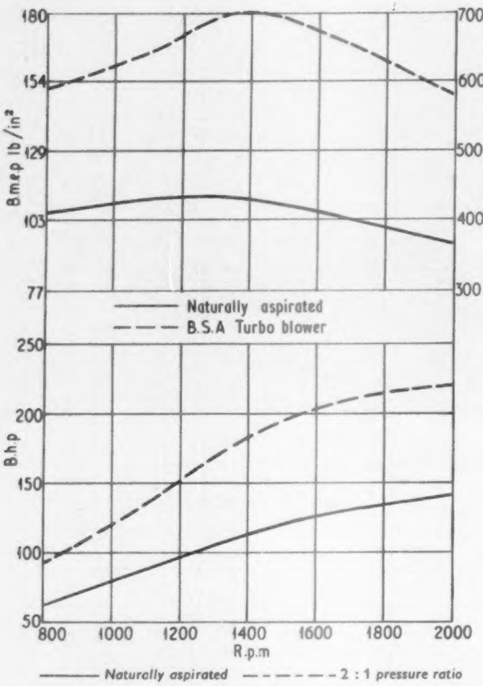


Fig. 25. Performance curves recorded for an engine operated in the naturally aspirated and turbocharged conditions

and the lowest one 450 ft. On route 4, the total length of which was 19·44 miles, the steepest gradient was 1 in 13, and the highest and lowest points 1,033 ft and 500 ft respectively. The results of these tests are given in Table I.

A four-week fuel consumption test had previously been completed on route 4. The results obtained, which, of course, applied to naturally aspirated vehicles, were 8·62, 8·55, 8·95, 8·41 and 8·84 m.p.g. respectively for five different vehicles, the first of which was the same bus that was used for comparison with the turbocharged vehicle. In view of the differences in fuel consumption obtained with the turbocharged bus, by comparison with the naturally aspirated one, it was decided to operate both vehicles non-stop over route 2, for a measured distance of 28·01 miles. The results obtained with the naturally aspirated vehicle were as follows: time taken 1·17 hr, average speed 24·0 m.p.h., fuel consumption 11·13 m.p.g., while the results obtained with the turbocharged engine were: time taken 1·15 hr, average speed 24·4 m.p.h., and fuel consumption 11·63 m.p.g.

The results of the complete series of tests show that the aim, which was at obtaining with an 8·6 litre engine the same performance as that of a 10·6 litre unit, has been attained. In fact, the turbo charged unit gave better results on acceleration tests from a standing start. These results were as shown in Table II.

Since the fuel consumption figures obtained with the turbocharged double-deck buses were not as good as had been expected, further tests were carried out on Sunrising Hill, Warwickshire, which is approximately 0·7 miles long with a maximum gradient of 1 in 6½. These showed that the injection equipment used was unsuitable for this application. The C.A.V. N type injection pump, with 9 mm elements and an all-speed governor had been employed, because it could readily be adjusted to regulate the fuel delivery curve and therefore the torque output. However, a marked improvement in fuel consumption was obtained without loss of performance when this equipment was replaced by the C.A.V. standard N type unit, with 8 mm elements, and the rack set to limit the smoke to the same value as that obtained with the earlier installation.

The results of fuel consumption tests at idling speeds are given in Table III.

These results indicated that, by employing the standard N type fuel pump, a saving of 2 pt/hr could be obtained while the engine is idling. Moreover, since the standard pump cuts off the fuel supply when the engine is operated under over-run conditions above idling speed, a further saving of 4 pt/hr was considered to be obtainable in the over-run condition of operation of the vehicle transmission.

TABLE I

Details	Naturally aspirated	Turbocharged	Naturally aspirated	Turbocharged
	Both vehicles idled in neutral at all stops		Both vehicles idled in gear at all stops	
Route 1:				
Number of stops made	92	90	92	92
Running time	94 min	96 min	93 min	92 min
Average speed	13·38 m.p.h.	13·1 m.p.h.	13·48 m.p.h.	13·62 m.p.h.
Fuel consumption	7·70 m.p.g.	7·35 m.p.g.	7·47 m.p.g. 7·27 m.p.g.*	6·98 m.p.g 7·62 m.p.g.*
	All vehicles idled in gear at all stops			
Route 2:				
Number of stops made	144	141	144	138
Running time	139 min	129 min	138 min	131 min
Average speed	14·64 m.p.h.	15·75 m.p.h.	14·75 m.p.h.	15·55 m.p.h.
Fuel consumption	7·65 m.p.g.	7·81 m.p.g.	8·16 m.p.g. 7·70 m.p.g.*	8·19 m.p.g. 8·52 m.p.g.*
Route 3:				
Number of stops made	94	94	94	94
Running time	74 min	75 min	83 min	80 min
Average speed	13·6 m.p.h.	13·6 m.p.h.	12·18 m.p.h.	12·61 m.p.h.
Fuel consumption	6·77 m.p.g.	7.26 m.p.g.	7·77 m.p.g.	8·06 m.p.g.
Route 4:				
Number of stops made			98	100
Running time			82 min	82 min
Average speed			14·2 m.p.h.	14·2 m.p.h.
Fuel consumption			8·54 m.p.g.	8·75 m.p.g.

*These figures were obtained after changing the dual pump to a standard "N" type without all-speed governor. The repeat tests were made by traversing route 1 six times and route 2 five times.

Engine performance

The thorough investigation of the problems of turbocharging that has been carried out on a variety of automotive diesel engines, as well as on the test rigs, has not only given valuable information in respect of the performance to be expected from turbocharged engines, but it has also ensured that any possible sources of structural failure, either in the turbocharger or the engine, have been brought to light.

While most of the engine testing has, of course, been aimed particularly at improvement of the efficiency of the installation, investigations have also been carried out into the conditions necessary for matching the engine and turbocharger. Throughout these tests, emphasis has been placed on the attainment of the greatest possible fuel economy without sacrifice with regard to power output.

In Fig. 23, performance curves for an engine turbocharged to 1½-2 atmospheres and matched for fuel economy and smoke-free operation are compared with those for the naturally aspirated engine. The fuel injection rate was increased, but no structural alterations were made to the engine, and the valve overlap was unchanged. It can be seen that the power increase obtained at the 2 : 1 pressure ratio is about 30 per cent over most of the speed range. The torque curve can be adjusted to a large extent to suit individual applications; in this case, the torque curves illustrated were chosen to meet the requirements of an automotive installation.

TABLE II

Gear changes (at driver's discretion)	Normally aspirated	Turbocharged
1st to 2nd gear	4½ sec	4 sec
2nd to 3rd gear	12 sec	12 sec
3rd to 4th gear	36 sec	25 sec
4th to 3rd gear	74 sec	50 sec
3rd to 4th gear	90 sec	62 sec
Finishing point	164 sec 163 sec*	162 sec 153 sec*
Final speed	20 m.p.h.	23 m.p.h.

*These figures were obtained from repeat tests using a standard N type pump without the all-speed governor.

TABLE III

	Gear	Engine r.p.m.	Fuel consumption pt/hr
All-speed governor with 9 mm elements	1st Neutral	370 500	5·45 2·18
Standard N type with 8 mm elements	1st Neutral	300 350	2·86 1·36

That the fuel economy of the engine has not been adversely effected is shown by the curves of specific consumption at maximum power output. A further indication of the fuel economy of the turbocharged installation is given in Fig. 19, which compares the consumption loops for the naturally aspirated and the 1·5 : 1 ratio turbocharged conditions. Although with the lower brake mean effective pressures at the lower speeds, the fuel consumption of the turbocharged version of the engine is not quite so favourable, this loss is more than counterbalanced by the improved consumption at the highest mean effective pressures and speeds.

It should be noted that the power outputs have been restricted, by limiting the b.m.e.p. to give smoke-free operation. Within this limitation, governing can be arranged to enable the optimum performance to be obtained throughout the running range. With regard to smoke, it is obvious that when the engine is suddenly called upon to give increased torque or speed, there inevitably will be a slight delay in the response of the turbocharger. Under these conditions, the turbocharger momentarily will be unable to supply enough air to burn the extra fuel injected into the cylinders. It is this that tends to cause exhaust smoke. Tests, carried out on a hill having a gradient of 1 in 6½, have indicated that the smoke in the exhaust of a turbocharged engine need be no more than that of a naturally aspirated unit, and that this does not involve any serious sacrifice in performance.

Performance curves for an engine turbocharged to 2 atmospheres and matched for maximum torque are given in Fig. 25. The curves in Fig. 21 compare the performance, with the fuel pump racks set at various positions, of a naturally aspirated, 9·6 litre engine with that of a turbocharged engine of the same capacity. In the turbocharged installation, an intercooler was employed to increase the density of the charge introduced into the engine cylinders.

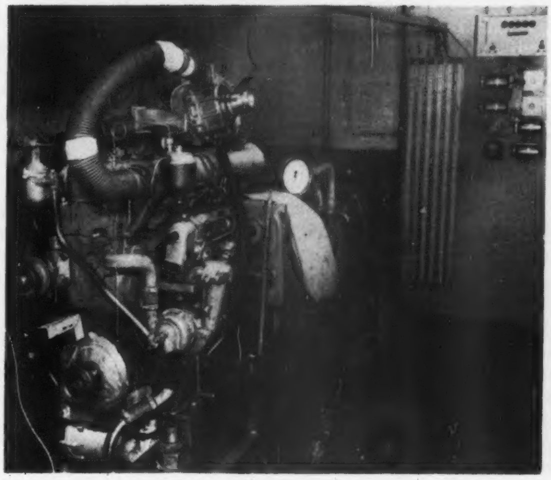
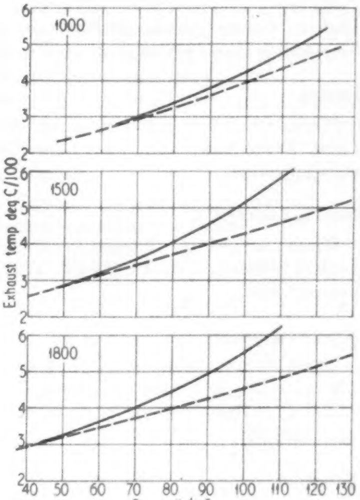


Fig. 26. The Daimler CD6 8·6 litre engine, equipped with a double-entry turbine for development tests. The instrumentation on the right includes a Decatron electronic counter used in conjunction with a photo-electric cell and a light beam reflected from a flat on the nut that secures the rotating assembly of the turbocharger



—— Naturally aspirated
- - - - Turbocharged

Fig. 27. Effect of turbocharging on engine exhaust temperature

Maximum cylinder pressures for a turbocharged and normally aspirated, Daimler CD6, 8·6 litre engine, are shown in Fig. 22. In this instance, the engine was turbocharged to 2 atmospheres. The effect of turbocharging on the exhaust temperature of this engine is shown in Fig. 27. Test installations are illustrated in Figs. 17, 18, and 26.

Production

Because of the high efficiency required with regard to aerodynamic performance, all the passages have to be smooth. Therefore, the B.S.A. turbocharger has been designed to eliminate costly polishing operations. To this end, pressure die castings are employed so far as possible for the aluminium components, and investment casting for components made of heat-resistant materials.

Compressor rotors, which are not so severely stressed as those used in turbines, are at present manufactured by gravity die casting. However, experiments are being carried out with rotors made by both the pressure and the vacuum die casting processes. The principal difficulty with pressure die casting is the avoidance of local porosity due to trapped air, a trouble obviated by vacuum die casting.

To enable a wide range of air flow capacities to be obtained, the diffuser ring is made separately from the compressor casing. It is a pressure die casting, inserted die forms being used so that a range of rings can be produced from a common die. The centre casing also is a pressure die casting. It is stiffened by a large number of thin section ribs.

Foundry technique plays an important part in the successful production of turbine discs, and the B.S.A. group of companies have had considerable experience of this work. Nozzle rings with cast iron back plates and separately-mounted, stainless steel blades are used for matching and other development work, but the production rings are precision castings of Jessops G.18B heat-resistant steel. They are made by the wax investment process, as also are the turbine rotors.

Because of the high length : diameter ratio of the rotors, they have to be dynamically balanced. Spigots are machined at each end of the rotor boss to locate it in a small balancing machine. After each component has been balanced separately, the whole assembly is checked. This final check rarely indicates any need for further adjustment. The tolerance that is specified for production is 4 milliounce-in.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

OCTOBER

London

Tuesday, 8*th October*, 6.0 *p.m.*, *at* 1 *Birdcage Walk, Westminster, S.W.*1. *Automobile Division General Meeting. Address by the Chairman of the Division.*

Birmingham

Tuesday, 29*th October*, 6.30 *p.m. The James Watt Memorial Institute, Great Charles Street, Birmingham. Paper: "Some Important Problems Concerning the Small Utility Car," by Dr. Ing. D. Giacosa (read by J. H. Pitchford, M.A., M.I.Mech.E.).*

Derby

Monday, 21*st October*, 7.15 *p.m. The Midland Hotel, Derby. Paper: "Some Problems in Lubrication of Small Two-Stroke Petrol Engines," by A. Towle, M.Sc., M.I.Mech.E.*

Luton

Wednesday, 9*th October*, 7.30 *p.m. Assembly Room, Luton Town Hall. Address by the Chairman of the Automobile Division.*

North-Eastern

Wednesday, 16*th October*, 7.30 *p.m. The Chemistry Lecture Theatre, The University, Leeds. Paper: to be announced later.*

North-Western

Monday, 14*th October*, 7.15 *p.m. Grosvenor Museum, Chester. Paper: "An Investigation into the Mechanism of Oil Loss Past Pistons," by P. de K. Dykes, Ph.D., M.A., M.I.Mech.E.*

Scottish

Monday, 21*st October*, 7.30 *p.m. The Institution of Engineers and Shipbuilders*, 39 *Elmbank Crescent, Glasgow, C.*2. *Address by the Chairman of the Automobile Division.*

Western

Thursday, 31*st October*, 6.45 *p.m. Royal Hotel, Bristol. Address by the Chairman of the Automobile Division.*

Coventry Graduates' Section

Wednesday, 16*th October*, 7.30 *p.m. The Wine Lodge, The Burges, Coventry. Joint meeting with the Midland Graduates. Paper: "Gas Turbines and Their Marine Application," by R. H. Ashton, B.Sc.(Eng.), G.I.Mech.E.*

NOVEMBER

London

Tuesday, 12*th November. Automobile Division General Meeting. Paper: "The Suspension of Internal Combustion Engines in Vehicles," by M. Horovitz, B.Sc.(Eng.), A.M.I.Mech.E.*

Coventry

Tuesday, 5*th November*, 7.15 *p.m. Grosvenor Room, Leofric Hotel, Coventry. Address by the Chairman of the Automobile Division, R. C. Cross (Member), entitled "Some Experiments with Internal Combustion Engines."*

Luton

Wednesday, 6*th November*, 7.30 *p.m. Assembly Room, Luton Town Hall. Paper: "Some Important Problems Concerning the Small Utility Car," by Dr. Ing. D. Giacosa (read by J. H. Pitchford, M.A., M.I.Mech.E.).*

North-Western

Tuesday, 12*th November*, 7.15 *p.m. Crewe Arms Hotel, Crewe. Paper: to be announced later.*

The Minimouse

An Argument Favouring the Development of Smaller Cars

GEORGE WANSBROUGH

THE leading articles of the *Automobile Engineer* are, as a rule, so well balanced, wise, and convincing that one is tempted to join issue when there is so much that seems disputable as there is in the September leading article which concludes that "there is no doubt that the arguments against are stronger than those for the development of a very small car in this country."

It is argued that "the British motorist is accustomed to a relatively smooth running, reasonably quiet engine with four or six cylinders; to get equivalent smoothness and quiet running from an air-cooled twin cylinder power circuit will not be easy." It will, indeed, not be easy to get such smoothness; but the *Autocar* of July 5th reports of the new Fiat 500 that "the outstanding impression is of complete engine smoothness under all conditions." But even if smoothness is sacrificed, the argument would not be conclusive; for the appeal of the very small car—let's call it the "Minimouse"—is not so much to the British motorist as he is, but to those who at present are not motorists, and can only become such if motoring is made very much cheaper: for them the comparison is not between, say, a Morris Minor and a Minimouse, but between a Minimouse and no motor car at all; and then a great deal of refinement can be sacrificed for the sake of the enlarged freedom of movement, the terrific widening of horizons, that the Minimouse will give. (If the accommodation, comfort and refinement of a Minimouse are compared with a motor cycle combination, the advantages of the former are prodigious.)

The key factor is the income level at which the ownership of cars is possible; and it is hard for those whose income is such that they take their ownership of a car more or less for granted to realize how ready the less affluent are to make sacrifices for the ownership of a car. An attempt has been made to correlate the ownership of cars and income levels on the basis of the number of cars in circulation. The figures are quite staggering. In September, 1955, there were registered in this country 3,478,950 cars. If one compares this with the number of incomes (*before* tax) shown by the Inland Revenue's last published report, which are those for the year ended April, 1955, and tries to make a graded allocation of that number of cars with the number of incomes at different levels, so as to estimate the proportion of car owners at the different levels of income, one has the following table:—

Income before Tax	Number of Incomes	Cumulative	Suggested Number and Proportion of Car Users in each Income Group		
	1,000s	1,000s	percentage	Number 1,000s	Cumulative 1,000s
Above £1,500	550	550	95	523	523
£1,500-£1,000	745	1,295	85	633	1,156
£999-900	475	1,770	70	332	1,488
£899-£800	855	2,605	55	459	1,947
£799-£700	1,470	4,075	45	662	2,609
£699-£600	2,140	5,215	20	428	3,037
£599-£500	2,800	9,015	10	280	3,317
£499-£400	3,320	12,355	5	166	3,483
	12,355			3,483	

It is clear from this table that, in 1955, whether or not the *proportions* shown above are close to the facts, car ownership *must* have penetrated deeply into the ranks of those with an income before tax of less than £800 a year; and equally, it is clear that there is a very substantial margin of people at about the same income levels who were then not car owners. In other words, there are in this country a tremendous number of potential car owners near the margin to whom, presumably, the last £5 a year of running cost is all important; and any substantial reduction in the cost of ownership can be expected to do much to open up that market.

Further light is thrown on the possibilities of extending car ownership by a study of car ownership in different parts of the country. The Inland Revenue gives a similar return of incomes before tax for the year 1949/50 (alas, nothing later than that is yet available) county by county; and building up for that year a scale of proportions—on the same lines as that above, but fitting the total car ownership in 1950, one finds that the differences between counties and districts are quite remarkable. For example, two districts, the West of England, and East Anglia, each had nearly twice as many cars as would have been given by applying to that district the scale which, applied to the incomes of the country as a whole, would give the actual number of cars in circulation in 1950. This shows that, when the case for a purchase is just a little stronger—greater distances to go for shopping, poorer public transport facilities and so on—then there is immediately a much greater readiness to buy a car, among people of identical income. In other words, there is what the economists call great "elasticity of demand"; so that the increased number of purchasers who will be drawn into the market by a comparatively modest reduction in price is usually great. If the Minimouse would reduce very substantially the lowest possible cost of motoring, say by 20-30 per cent, the increase in the number of potential car owners would be enormous, and the market for the vehicle very great.

It is clear, then, from these figures that there is an enormous potential car-market to be opened up at lower prices than the minimum costs of cars of the smallest size at present made in this country. What are the possibilities of offering, by scaling down, very much cheaper motoring ?

The leading article refers to a reduction in original cost of 20 per cent achieved by a reduction in weight of 20 per

cent. It is probable that, with a similar volume of production, it is not quite possible to achieve with a smaller vehicle quite the same low cost in d. per lb., unless, as your article suggests, some sacrifice in trim, finish and instrumentation—i.e., a sacrifice of some components of a higher cost in d. per lb—is accepted. But a saving of more than 20 per cent in weight and in price seems to be possible. The following table sets out the kerb weights of the leading small European cars (other than the Minimouse class) and, in the case of the English cars, their retail prices excluding P.T.:—

	Kerb Weight cwt.	Retail Price	
		£	d. per lb
Morris Minor	15½	416	57·6
Ford Anglia	15¼	380	53·4
Austin A.35	14	379	55·8
Renault 4 c.v.	11¾		
Fiat 600	11¼		
Citroen 2 c.v.	10½		

The Fiat 600 is already, as near as may be, 20 per cent less than the kerb weight of the Austin A.35, and the Citroen 2 c.v. 25 per cent less. The Fiat 500, going to a 480 c.c. air-cooled twin, appears to have a kerb weight of 9½ cwts. But if a still smaller engine is accepted, there is the Goggomobil, with an engine of 293 c.c. and a kerb weight of 7⅞ cwt. The last named has accommodation that will quite adequately meet the needs of a small family—parents and two children—and a quite acceptable standard of performance and comfort. According to the *Autocar's* Road Test, the Goggo saloon on test gave 58 m.p.g., against 41·3 for the Austin A.35.

It seems reasonable to suppose that any of our major car producers could produce a car of say 400 c.c. with just a little more accommodation than the Goggo, at a kerb weight of say 9 cwt; and allowing for a slightly higher price in d. per lb than those set down above, say 60d. per lb, giving a price of £252 plus tax, a saving in first cost compared to the A.35 of fully one third. The saving in fuel costs would probably be of the same order. It would seem conservative to say that the total costs of ownership would be no more than three-quarters of the ownership of the Morris Minor, Ford Anglia or A.35; and on the table of incomes above, one might well look for a market in this country which would at least justify planning on output rates that would equal those planned—so it is said—for the Fiat 500, of 600 a day or, for the Goggo, 400 a day; and, it is easy to believe, figures substantially higher even than these.

There would, of course, be a very big investment to make in manufacturing plant; not only in the works of the manufacturer himself, but also of the suppliers; for if the lowest possible weight and cost were to be achieved, most of the components—electrical equipment, carburettors, wheels, and what have you—would need to be smaller than those at present in high production, and in many cases—for lowest cost—special manufacturing plant and equipment would be called for.

Your article questions the desirability of making over to such a vehicle any of the existing assembly lines; defending the British manufacturers against the charge that they produce too many models. It is surely, is it not, the investment in machinery equipment, in plant for making components prior to the assembly lines, that constitutes the major investment. But apart from that, while your defence is valid against a great deal of mistaken criticism of the industry, it does not seem to meet the more justified criticism, namely that the British industry would do better in competition with the European manufacturers if there was in this country some single model, or better still, two or even three models, so outstandingly attractive to the customer as to justify the high rates of production achieved by the V.W. (1,800 units a day) or the Renault Dauphine (probably 800 a day).

One way perhaps in which the problem might be tackled would be for one of the big manufacturers to give up one of his existing production lines, in order to make a Minimouse, and drop one size of model, in agreement with the other major manufacturers that they would, say for a period of three years, leave him a clear run in the Minimouse market.

The Minimouse type of car may or may not be the answer; but it is hoped that this article gives valid reason for suggesting that the case is stronger than your leading article would allow. This country gave the world the best early example of a Minimouse in the Austin 7 whose original kerb weight, in saloon form, be it remembered, was less than 7 cwt. It would be a great pity if it was to leave this market in the 60's to its Continental competitors.

(At one point in developing his argument, Mr. Wansbrough is less than just to us—we referred to production and not merely assembly lines.—Ed.)

POINTS ABOUT DIECASTING

An interesting booklet "Diecastings—Your Questions" has been produced by Fry's Diecastings Ltd. In a short compass it contains a considerable amount of useful information, from which these notes are taken. Complimentary copies can be obtained on application to Fry's Diecastings Ltd., Midland Works, Brierley Hill Road, Wordsley, nr. Stourbridge.

Among the widely used discasting alloys, zinc alloy pressure diecastings provide the nearest approach to the finished product. Small holes down to 0·05 in diameter can be included; lettering or decorative features can be cast; all this with close dimensional accuracy and good surface appearance. Aluminium or magnesium alloy pressure diecastings come next for accuracy. In these castings holes of under 0·10 in diameter are usually omitted. Aluminium alloy gravity diecastings do not provide quite so close an approach to the finished product as do pressure diecastings. Even so holes of 0·125 in and upwards can often be included. It is preferable to drill holes under about 0·15 in diameter in brass or aluminium bronze diecastings.

It is not possible to give all-embracing figures for limits of accuracy, because so much depends upon the design of the component. In addition, a dimension which is cut by the parting line of the die could not be expected to be as accurate as a dimension bounded by a single die block. For a given dimension in identical components the following comparisons will serve to indicate the approximate ratio between the accuracies of the various alloys in the diecast condition:—

Zinc alloy, pressure diecast	±0·0015 in
Aluminium or magnesium alloy, pressure diecast	±0·003 in
Aluminium alloy, gravity diecast	±0·007 in
Copper alloy, gravity diecast	±0·010 in

In planning a new diecasting, the assembly should be considered as a whole. It frequently happens that an article previously fabricated from several components may be made as a one-piece diecasting.

The Locking of Nuts

A Discussion on How Far Anything More Than Normal Tightening is Required

S. H. Grylls

It is not unreasonable to ask whether any nut requires locking, where the term "locking" means a separate operation from the normal tightening of an ordinary nut that consists of a continuous thread in a homogeneous material of constant section. These notes are based on investigations, over several years, into this question. The examples are mostly taken from automobile engineering, but the conclusions drawn may be of more general application.

Nuts and bolts or set screws are mostly used to hold two or more things together, for example, the attachment of a shock absorber to a frame. Less frequently they are used to stop things from coming apart, for example, a pair of scissors. These latter applications are the more troublesome; a screw thread is often used for convenience, and the thread is not necessarily stretched and riveting might have served instead.

These notes are almost wholly concerned with applications in which a nut has been used to stretch a bolt, compressing whatever there is between the nut and the bolt head. If the maximum applied forces on such an assembly are insufficient to "fidget" the structure, then it is the author's contention that locking is not required. (The word "fidget" is used throughout to denote small relative movements between adjacent pieces.) If this theory is accepted, it is still necessary to consider whether any contrary evidence is forthcoming, whether it is always possible to assess the maximum loads imposed, and whether any quite different motive for locking can arise. In the author's experience no contrary evidence has come to light, although the effects of loads beyond those predicted have been observed on a number of occasions. There is, however, a valid, but quite indirect, reason for a separate operation to lock some nuts.

Why is it common practice to lock (usually by split pins) the connecting rod bolts of an automobile engine, but to take no care of the cylinder head studs? If anything, the latter, due to differential expansions, are more likely to come loose. Two possible answers are: "Because they always have been locked" and "Imagine the castastrophe if they came loose!" Thousands of hours of test bed running and thousands of miles on the road with small petrol engines, without a sign of a nut coming loose, have proved that locked nuts are not necessary.

However, the connecting rod nut is a vital nut. The stress in the bolt due to spannering must exceed the maximum inertia loads of high speed running. This applies to every nut of every connecting rod. A split pin overcomes what may be called the "lunch time problem," when work is left in an uncertain state of completion. No fitter will lock a nut unless he has just made sure it is tight—split pinning provides this inspection and avoids the danger that some nuts may be left loose. A connecting rod big end nut is locked for psychological, rather than technical reasons.

As examples where locking is technically justified are discussed, it may be as well at this point to consider, from the academic point of view, the value of the various ways of locking a nut. The accompanying table shows the results of a series of tests in which $\frac{3}{8}$ in B.S.F. nuts were used. Both nut and bolt were of unplated steel. The undoing torque, measured by torque spanner, was determined immediately after tightening. The conclusions drawn from these static tests are:—

(1) The split pin when sheared provides a 30 per cent increase in torque. (The torque to shear a split pin cannot all be added to the normal undoing torque unless the pin is tight in its hole. Nuts must never be undone to line up the split pin hole; split pins are therefore not easy to use on very short bolts.)
(2) Some locks are worse than nothing.
(3) Most locks provide about 10 per cent increase of torque.

Although it is rather a digression from the actual problem of locking, it is interesting to consider why a plain nut

UNDOING TORQUE IN LB-IN OF NUTS—$\frac{3}{8}$ in B.S.F. UNPLATED

All nuts initially tightened to 400 lb-in clamping 0·104 in sheet steel of 108 Brinell

	Single full nut	Spring washer	Grover washer	Castellated nut split-pinned Finger-tight	shake-proof washer	Standard full nut reduced facing $\frac{1}{2}$ in diameter	Special nut 1·200 in dia. with 1·200 in washer under bolt	Self-locking nut (a)	Self-locking nut (b)	Self-locking nut (c)	Self-locking nut (d)
1	310	340	260	100	290	220	310	340	300	340	300
2	290	350	300	90	250	200	320	320	280	340	300
3	310	330	320	110	240	200	310	330	310	320	290
4	300	340	340	120	260	190	300	330	300	310	280
5	320	330	320	110	290	210	290	320	290	280	280
6	260	360	250	120	260	210	300	300	300	300	290
7	310	220	220	100	270	220	300	320	300	300	300
8	300	310	280	110	280	210	280	330	310	280	310
9	280	310	340	100	240	180	300	310	310	300	280
10	290	310	300	120	260	190	310	320	300	300	280
11	320	280	260	110	240	200	270	320	280	320	280
12	300	260	280	110	270	190	280	340	300	300	290
Aver.	299·1	317·5	289·1	108·3	262·5	201·7	289·1	323·3	298·3	307·5	290

undoes at 75 per cent of the tightening torque. A simple start to explain the 75 per cent would be:

If E is the load in the bolt and the coefficient of friction on both the thread and the face is 0·10 (a value derived from knowledge of the torque required to fail a stud by continued tightening), then the torque to overcome face friction is:

$$Tf = E \times 0{\cdot}10 \times 0{\cdot}244$$
$$= 0{\cdot}0244E$$

and the torque to overcome thread friction is:

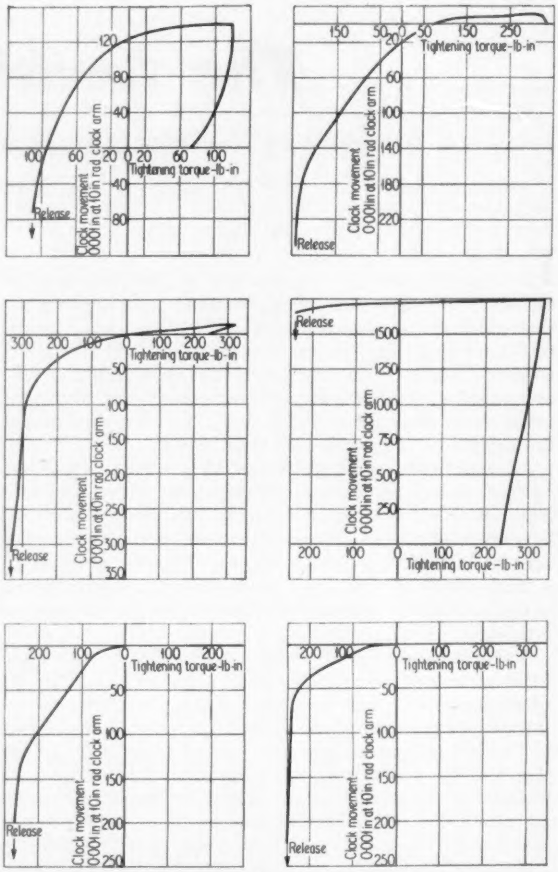
$$Tt = E \times 0{\cdot}10 \times 0{\cdot}174 \times 1{\cdot}0/0{\cdot}866$$
$$= 0{\cdot}021E$$

Owing to the angle of thread there is a torque helping or hindering of E/125.

When a nut is tightened, the bolt is left twisted and it is reasonable to expect an initial backward movement when only the face slips at 0·47 of the tightening torque. This would be followed by complete slippage at a negative torque about 30 per cent less than the tightening torque. As torque spanner results did not show the first part of this result, the apparatus shown in Fig. 1 was constructed to measure torques and movement more accurately. It was hoped that by plotting an hysteresis loop, the sequence of events could be observed at various discontinuities in the curve.

Fig. 2 shows a series of loops under different conditions. The curves are chiefly of interest in showing the effect of cadmium, which is now frequently used. They also show that a nut subjected to a continuous steady torque rotates slowly for hours or even days. A nut tightened in this manner requires a larger undoing torque, due, presumably, to the rise in the coefficient friction through the gradual elimination of lubricant. From this it can be deduced that nuts are less likely to come loose as time goes on, a fact often noticed. It is difficult to explain the shapes of the hysteresis loops; the expected discontinuity is not present in any case, and no theories to explain this have presented themselves within the scope of this article.

Here it is appropriate to mention the worst way of attempting to lock a nut, namely, by riveting over the end of the bolt. Experiments on the fixed end of a wheel stud with a $\frac{1}{2}$ in thread produced the following results. An average initial torque of 65 lb-ft fell to 55 lb-ft after riveting. On undoing the nuts the torque fell to 15 lb-ft before the



Top Left—Bright nut and bolt bearing on frame material of 108 Brinell, surface enamelled. Duration of test $1\frac{1}{2}$ hours
Top right—Cadmium plated bolt and nut loaded to 336 lb-in without appreciable delay between load increments. Stretch of bolt 0·0017 in. Unloading started after five minutes
Middle left—Unplated bolt and nut bearing on steel surface of 122 Brinell. Initially tightened with a torque of 240 lb-in and left loaded for 14 hours before readings were taken
Middle right—Cadmium plated bolt and nut tightened to 240 lb-in and left loaded for 15 hours, followed by additional loading at intervals from $1\frac{1}{4}$ to $2\frac{1}{4}$ hours
Bottom left—Cadmium plated bolt and nut tightened without delay between load increments to a final torque of 336 lb-in. Stretch of bolt 0·0007 in. Left loaded for seven days
Bottom right—Cadmium plated nut on unplated bolt tightened with a torque of 336 lb-in. Stretch of bolt 0·0003 in. Left loaded for seven days

Fig. 2. Hysteresis loops for various conditions with $\frac{3}{8}$ in. B.S.F. nuts

Fig. 1. Apparatus for measuring undoing torque

riveting exerted any influence, after which the torque rose to 45 lb-ft in order to shear the metal.

It is the writer's contention that a correctly used nut cannot *shake* loose. Of course, the expression "correctly used" includes "correctly tightened". Consider the case of a $\frac{3}{8}$ in nut with an undoing torque of 300 lb-in. To loosen such a nut, applied accelerations are required of magnitude rotationally of one thousand million radians/sec^2 or linearly of 5,000,000g. It is unlikely that such high figures are ever encountered. For example, unusually high rotational acceleration is experienced during the critical speed of a six-cylinder crankshaft, but ± 3 deg at 5,000 r.p.m. represents an acceleration of only 125,000 radians/sec^2. Furthermore, provided the nut is retaining a part of greater inertia than itself, the part will shift at a lower acceleration than the nut. Such a shift or fidget predicates incorrect use of the nut.

Nevertheless, it must be admitted that there are conditions that call for some form of locking. They are: (a) where it is impossible to prevent the structure from fidgeting, and (b) where it is not economical to cater without fidget for very occasional overloads. This latter condition applies to a number of components in the assembly of an automobile. For example, the bolting of an engine mounting bracket to

the frame should not fidget up to $3g$, the friction at any joint being alone sufficient for this duty. Beyond $3g$ at the occasional load due to unexpectedly bad road surface ($6g$ is sometimes encountered) the joints will slip and the shear load on the bolts will take some of the load. On each of these unusual occasions some wear takes place and an undoing torque may be applied to the nuts. Split pinning of the nuts will increase the life of the car at a smaller cost than the provision of more and larger bolts. It is not safe to rely on a coefficient of friction of more than 0·1 between dry mating surfaces and, to ensure an infinite life of a structure, sufficient bolting must be used to prevent fidgeting under the maximum conditions ever encountered, even on one single occasion.

There is a class of joint between those that are to be firmly held together and those of the scissors type. For this class locking is essential. Generally such a structure involves the compression of a soft material such as rubber. The stress in the securing bolts will be low and, therefore, the undoing torque of the nuts will also be low. Most car rear axles have a rubber sandwich between themselves and the springs. The load on the U-bolts is only about one-half of what they can carry. Here locking of the nuts is essential, not only in name but by a method that provides adequate friction resistance. A second nut tightened against the first is about the only means of successful locking. There are very few other examples in automobile engineering where what is commonly called a "lock nut" is the correct practice.

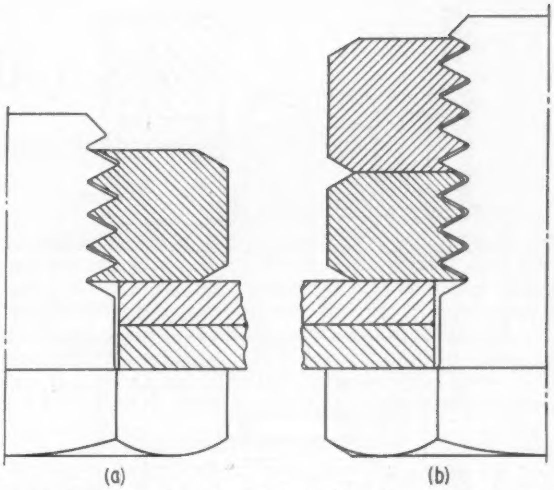
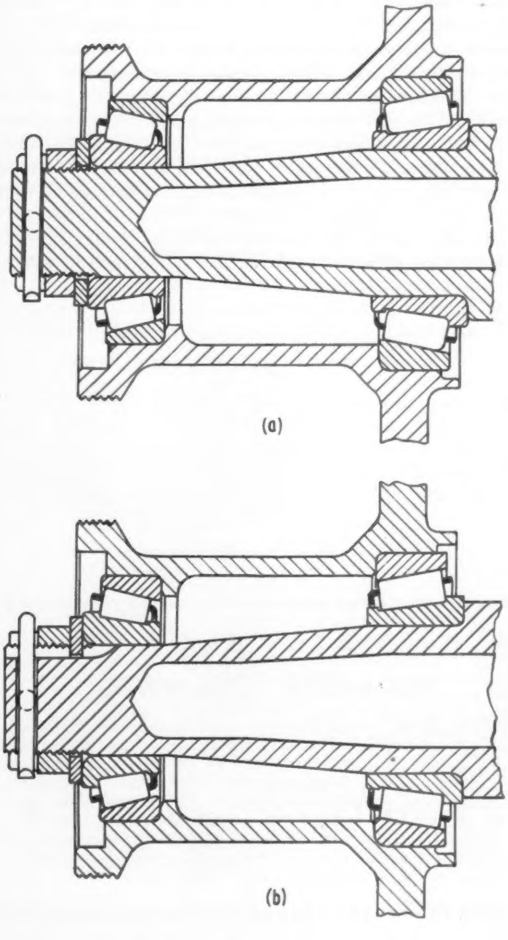


Fig. 3. Comparison of a single nut and a lock nut; there is thread contact only on the top nut

Fig. 4. Methods of securing front hub journal bearings



Strangely enough, a soft cork washer is normally fitted between the engine sump and the cylinder block but the nuts do not slacken and finally come off. Vibration must cause very little relative motion in this case.

There is no harm in reiterating the known, but seldom acted upon, truth that in a firm joint the second nut to be put on takes the load and should be the thicker of the two. If the threads have the usual slackness, the first nut is relieved of the load and becomes a distance piece. However, a greater length of bolt has been stretched, and this permits a larger allowance for wear of parts that occasionally fidget, Fig. 3.

Spring washers are used with this same intention, but their effect does not withstand mathematical investigation. The load to flatten a spring washer is about one per cent of the tension in the bolt whose locking it is supposed to be assisting—a common use would be under the nut of a bolt stretched 0·001 in to 6,000 lb tension. After 0·001 in wear due to fidgeting, the tension in the assembly will drop from 6,000 lb to 70 lb, a useless figure. Some tests on a structure shaken with just sufficient acceleration to fidget the joint showed that the nuts loosened more quickly when spring washers were fitted than when plain washers were used. A simple explanation was found in that spring washers are not flat when compressed. The best "lock" proved to be a flat washer of at least 200 B.H.N.

There are many applications where a nut is used to secure a moving part on a journal bearing. For example, the front hub of a motor car is a problem of journal bearings; Fig. 4 shows orthodox designs. The inner race of the outer bearing is likely to creep and in so doing can apply quite a large friction torque on the nut. Since a slight swashing motion can easily exist owing to slacks or imperfections in machining, the torque is unlikely to be a pure torque. It is generally accepted that the inner race creeps forward and never backwards. As nearly all motoring is in a forward direction, it is possible by fitting a left-hand nut on the near side of the car, to arrange that the creep of the inner race will tighten the nut. This arrangement is satisfactory if further tightening of the nut will not further pre-load the bearings. If the nut is tightened against a shoulder and shim adjustment of bearing pre-load or slack is used, the design is satisfactory. Generally however, the nut is used to set the pre-load or slack, and is then split pinned. In this case the split pin must be large enough to resist in shear the creep torque. It can be relieved of this torque by the insertion of a splined washer between

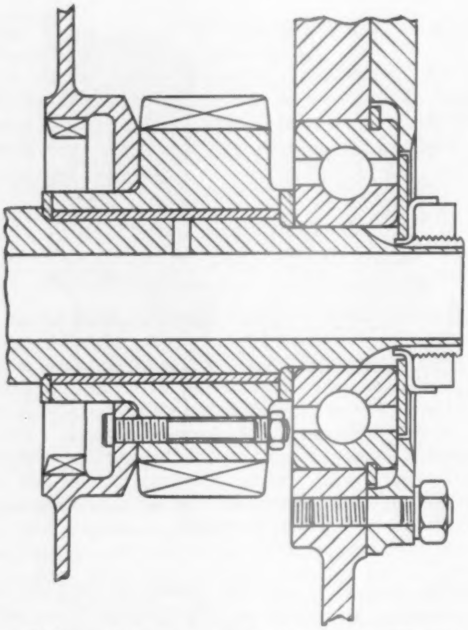
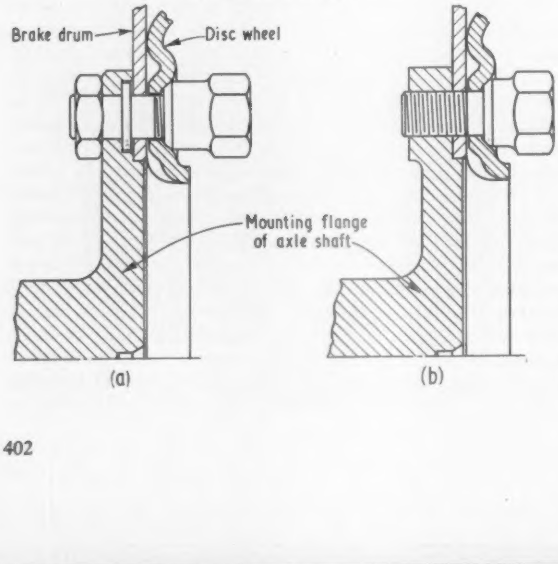


Fig. 5. This loose helical gear application created difficulties, and extreme spannering proved to be the only effective method of retention

the bearing and the nut, a construction that is shown in (b) at Fig. 4. The use of handed threads would ensure that the road wheels do not come off, should the locking devices fail or be accidentally omitted.

A more difficult example of the same fundamental problem is the retention of a loose helical gear. An application that gave an immense amount of trouble is shown in Fig. 5. The gear is in the second speed train of a motor car. Where any ratio other than second is employed, the pinion rotates freely on the shaft and all is well. When second speed is engaged the pinion is made to rotate with the shaft by engagement of the dogs. As far as its journal clearance will allow, the pinion runs tilted, and it also fidgets on the shaft in an epicyclic manner. The tooth load was about 2,000 lb, and the nut had to withstand a large torque combined with a large swash load. No locking device would keep the nut tight, and the problem resolved itself, as is often the case, into a matter of using as much brute force as possible in the restricted dimensions available.

Fig. 6. Methods of attaching road wheels by studs and nuts at (a) and by set screws at (b)



Road wheel retention on automobiles raises some interesting points. The wheel fixing is subject to a large journal load, on which is superimposed the variations due to the accelerations from road surfaces, a thrust load reaching to about half the journal load and applied offset from the axis, and, lastly, the torque of driving or braking. That the wheel is not infinitely rigid provides a fourth consideration to complicate the problem still further.

Every car today has enough nuts of sufficient size and at such a radius to ensure that if the nuts are really tight, trouble will not occur. However, the fitting of a road wheel is not always in the hands of a skilled fitter. The spare wheel may be fitted by a woman after a puncture, and it may be so thick in paint that even when securely fitted the bolt tension soon disappears. Wheels always fidget and the paint film soon vanishes. The fidgeting of the wheel may be described as a mixture of axial movement, a rotating D-shaped distortion always at the bottom, and an epicyclic creep. The last of these produces a torque on the spherical seat of the nut, and, if as is common practice right-hand threads are fitted all round, the torque is in the loosening direction on the near-side. The reason why all near-side wheels do not fall off must be that there is sufficient tightness to prevent the epicyclic creep, and only the other two motions take place. In other words, fidgeting is local and always at the bottom.

If it is desired to cater for wheels that are not fully tightened, left-hand threads must be used on the near-side. Experiments proved this point in a few miles, but left apparently unsolved the question why nuts sometimes disappear on a car with correctly handed threads. Consider the off-side; although the nut seat of a right-hand nut produces a tightening torque, the nut thread produces a loosening torque. If a nut becomes so loose that it no longer ever touches the wheel, it will finally spin off under epicyclic movement and inertia forces. Apparently the transition from seat control to thread control must be due to one nut being looser than the others. Unlike some applications quoted in these notes, the wheel nut stays tight if once tight, but benefits from a form of locking if left loose. In most other instances there is no form of locking that benefits a loose nut.

Since the theme throughout has been that locking is generally unnecessary, an examination of the costs of various common arrangements should not come amiss. Using ⅜ in B.S.F. as the standard, it has been found that the cost per 1,000, including, where appropriate, the extra labour involved, was:

	£	s.	d.
Nuts alone in En 6 material	4	4	0
Nuts split pinned	32	13	0
Nuts locked in pairs by double tab washers	12	11	6
Nuts locked by single tab washers	13	15	0
Proprietary self locking nut	6	0	0

Micronic Dry Filter

A New Air Filter with a Phenolic Resin Impregnated Paper Element

RECENTLY, a new range of air filters has been introduced by the Purolator Filtration Division of Automotive Products Co. Ltd., of Leamington Spa. These filters have been designed for internal combustion engine air intakes, and can be used with or without a centrifugal type pre-cleaner. The units are available in a number of standard sizes, given in the accompanying table, but special filters can be produced to suit the requirements of individual engine manufacturers.

The new range, termed the Micronic Dry Type Filters, is a development from the well known Micronic oil filters. Each of the units comprises simply a conical base plate and the impregnated paper filter element, over which is fitted an inverted cupped cowl, or housing. The whole assembly is held together by a single tie bolt. Air enters the unit through an annular space between the lower end, or rim, of the cowl and the outer periphery of the conical base plate. It then passes through the filter element and out, by way of a hole in the centre of the base plate, into the engine air intake.

In detail, the construction of the filter is as follows. A sleeve adaptor to fit over the engine air intake is incorporated round the central hole in the pressed steel conical base plate. The periphery of the base plate is flanged to receive a cork or rubber washer, on which seats the filter element. This element is cylindrical in form, and is specially designed to withstand even the most severe loads that might be imposed on it, for example, if the engine backfires. It is made from a phenolic resin impregnated strip of thick absorbent paper, stiffened by longitudinal ribs, and folded concertina fashion. The ends of the strip are joined by a steel clip to form a cylinder, which is retained between two concentric, perforated steel sleeves. These sleeves considerably strengthen the filtering element.

Both ends of the element are dipped in a sealing compound, by means of which they are stuck to two inwardly-lipped, annular end plates of hardboard. The pressed steel cowl seats on top of the filter element, a cork or rubber sealing washer being interposed between the two. A captive nut is carried in a hole in the centre of the top of the shroud, and therefore cannot be lost. It is screwed on to the steel tie bolt that extends up from the conical base plate.

Of the noteworthy features of the unit, the most important is the use of a plastics-impregnated paper element, which arrests the minute particles of dust on its surface instead of in its pores, as does a felt element. Therefore, there is less danger that they might subsequently work their way right through. This is because the plastics-impregnation stabilizes the pore size and provides strength and rigidity, so that there is no possibility of the air pressure opening up blocked pores and forcing trapped particles through.

STANDARD FILTER SIZES

The nominal capacity is based on a pressure drop of $2\frac{1}{2}$—3 in W.G. across the filter.

Type	Nominal capacity C.F.M.	Element size, diameter × length	Connection* size
MF-19100	50	$5\frac{1}{2}$ in × $2\frac{5}{16}$ in	$1\frac{1}{2}$ in inside diameter
MF-19000	100	$5\frac{1}{2}$ in × $4\frac{9}{16}$ in	$1\frac{1}{2}$ in ,, ,,
MF-19201	200	$8\frac{9}{32}$ in × 4 in	2 in ,, ,,
MF-18400	350	8 in × $7\frac{9}{16}$ in	3 in outside diameter
MF-18100	500	$10\frac{3}{16}$ in × $11\frac{15}{16}$ in	4 in ,, ,,

* *Additional sizes are available.*

Another advantage of this form of filtration is that engine vibration tends to shake the heavier particles off the surface of the element. Nevertheless, it is still necessary to clean the element periodically. This can be done by blowing air from a normal pressure supply system through it in the reverse direction to the normal flow. Care should be taken to ensure that this air is free from oil or water contamination. Elements that have been cleaned several times in this way will finally clog and the air flow to the engine will become restricted. It is then necessary to fit a replacement. The elements should never be cleaned in any liquid, nor, of course, should engine breathers be piped into the cowl. Filtration becomes more efficient as dirt builds up on the element. If the unit is allowed to become badly choked, the engine will run rich and thus call attention to the fact that the element should be cleaned.

Other good features of the design are its light weight, simplicity and low cost. The manufacturers state that the filter is equally effective at all engine speeds, and that it is more efficient than the oil bath or oil wetted types. Since oil is not used in this filter, there is no possibility of oil being drawn into the engine. In addition, the unit is compact and can be installed at any angle. Servicing does not have to be carried out so frequently as with the oil bath or oil wetted types of filter, and the element can be cleaned quickly and easily. This latter feature, together with the fact that efficient filtration means less frequent engine overhauls, reduces operating costs. Restriction to the air flow by the Micronic filter is said to be negligible and therefore to have no adverse effect on fuel consumption and power output.

Noteworthy features of the new Purolator Micronic air filter are its efficiency, simplicity, light weight and robustness. As can be seen from this illustration, it comprises only three main assemblies, which are held together by a single nut. The nut is held captive in the top of the cowl and therefore is unlikely to be lost

CONTROLLED BARREL FINISHING

Process Record System Enables Specified Performance to be Immediately and Precisely Repeated

WHILE recorded history of the use of a barrel or drum for the polishing of metal parts extends back for about 170 years, in all probability the method is of much older origin. In Nature, the effect on stones on the seashore cascaded by wave action can hardly have escaped the attention of the not-unobservant ancients. Development of the process in the modern sense, however, is confined to the last 100 years. In 1857 a patent was granted for a barrelling machine to deburr "washers and the like" and between the years 1897 and 1914 more than fifty patents were granted. In short, improvement in the process was sought as the rate of component production increased and as manual labour time became more costly. Often the original aim of such improvement was to meet the requirements of a particular trade or a particular product but always in the background was the desire to render the process more predictable, more repeatable, more consistent in performance, and less dependent upon the acquired skill of the operator. Progress in recent times, roughly since the termination of World War I, has been rapid and today large, multi-unit, automatic-transfer, barrelling equipment can be integrated in the production lines of the most advanced manufacturing plants.

Possibly the outstanding item contributing to this end was the advent of synthetic media for inclusion with the work in the barrel. When natural media, such as granite chips, limestone chips, pebbles, gravel, sand, wood chips, and leather cuttings, were employed with water, oil or kerosine, the operator's experience was of major importance and each fresh supply of media had to be re-assessed. The synthetic material, mostly aluminium oxide, is fused at high temperature, broken down in a jaw-type crusher, screened for size, treated to remove the initial depreciation, washed, and finally graded. Thus, as it reaches the machine for use it is of specified characteristics as regards composition, size, density, cutting performance and depreciation rate. It is consistent throughout the batch and from batch to batch and can be selected for use and re-ordered by a code number.

This medium-capacity machine, model DB-50, has a barrel 22 in diameter and 24 in long rotating at speeds from 10 to 30 rev/min. Maximum load of parts and media is 800 lb

Very small parts of simple shape can be self-tumbled but, apart from such items, practically all parts require abrasive media in the barrel to effect some surface improvement or line removal before finally burnishing or polishing. This applies generally to castings, die-castings, forgings, stampings, pressings, and machined parts. The functions of the media are to:

1. Separate the parts and prevent impingement and consequent damage
2. Give bulk and weight to the mass in the barrel so that the pressure contact is adequate to cut or burnish the parts
3. Carry abrasive or polishing solutions to recessed or inaccessible areas of the parts
4. To impart lustre by high pressure contact in combination with lubricating or soap solutions.

In synthetic media the crystal growth and arrangement is controlled during production to ensure that, as the chip depreciates in use, new cutting faces are continually presented to the work. Cutting action is thus available throughout the life of the chips and, in many instances, the use of the more expensive abrasive-based compounds is not necessary.

Standard grades of media

The Almco Supersheen Division of Great Britain Ltd., Bury Mead Works, Hitchin, Herts., has standardized the production and supply of four different media to cover all barrelling requirements. For the fastest cutting action to remove burrs from parts of relatively simple form and to grind down the surfaces of castings and forgings, "Rapicut oxide" has been developed. This contains 99 per cent aluminium oxide, with consistent large grain size, structure and density throughout. By reason of its consistent cutting action and process time it is most useful for continuous production processes.

"Standard oxide" has a content of 98 per cent close-grained aluminium oxide. This also has a high cutting rate, coupled with a low rate of depreciation, and gives a matt finish to the work. It is particularly suitable for processing, without fear of chip lodgement, parts formed with holes or slots.

The "Standard bonded" chip is a compromise to give relatively fast cutting and finishing while securing a substantial economy in working costs. It contains 15 per cent aluminium oxide, 65 per cent silica, and the remainder various oxides. A feature of this medium is its ability to bring out the natural colour of the metal component being treated.

The "Super bonded" chip contains 55 per cent aluminium oxide and 45 per cent silica. It is very closely bonded and

gives a good cutting rate in conjunction with a very slow rate of depreciation. For fast cutting it can be used with an abrasive compound, which subsequently can be washed out and replaced with a polishing compound without unloading the contents of the barrel.

Of these four media, the three first mentioned are fused at temperatures up to approximately 1,100 deg C and are produced in Britain. The "Super bonded" medium requires to be fused at 1,900-2,000 deg C and for this reason is produced in the U.S.A. and is imported from the parent Almco organization. On this account it is somewhat more expensive than the other grades.

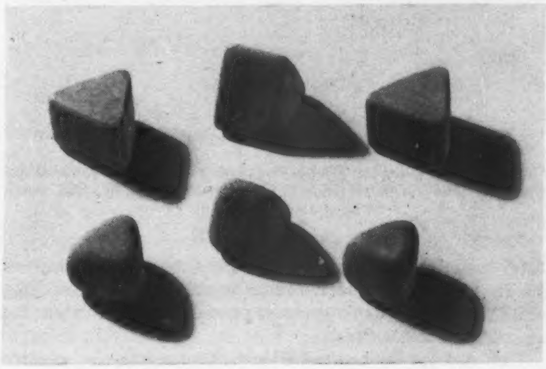
Steel diagonals

Such media are of random shape and a recent innovation by Almco Supersheen is the production of pre-shaped media. This is extruded in a continuous length of a particular section—circular, rectangular, triangular, diamond, or elliptical—and cut to specified lengths before fusing. Pre-shaped chips confer a further facility to the process, as the most suitable shape can be selected to suit the contours of the part to be treated. After continued use all chips tend to become more pebble-like but it is a feature of the pre-shaped chip to retain its basic shape for a considerable period. This characteristic is shown in the illustration of the pre-shaped triangular media. Considerable development has been undertaken on the manufacturing of shaped media. Pre-shaped chips are now available in three sizes of triangular form for deburring and grinding. The development of a polishing chip is now nearing completion.

Alternative to the synthetic media, steel diagonals or shapes are used for either grinding or burnishing. In a soft condition, steel diagonals are used in conjunction with an abrasive compound in the so-called "slug grinding" process for maximum metal removal in the minimum time. The compound provides the cutting action and the steel diagonals supply weight and keep the parts separated while serving to carry the abrasive material into masked or recessed areas of the work. After grinding with steel diagonals, the compound is washed out through a standard rinsing door and the run continued with fresh water and a burnishing compound. In this manner, a burnished finish is obtainable without the necessity of unloading the entire contents of the barrel.

More effective burnishing, however, can be obtained by the use of the special hardened steel shapes. These are formed to shape, ground, polished and then heat-treated to a glass hardness. If used solely for burnishing, they are guaranteed not to lose weight or shape and thus, in continuous production processes, obviate the necessity for repeated replenishing and resizing of media in order to prevent the lodgment of worm media in holes or recesses in the work. Suitably shaped "diagonals"—circular or angled sections cut to length at specific angles—can reach crevices, angles or grooves that cannot be approached by the steel balls, which are suitable for only relatively simple and straightforward work.

The weight ratio of diagonals to work will depend on the work to be processed and may vary from 2 : 1 to 4 : 1. In all cases where a highly burnished finish is required, it is necessary that barrel, diagonals, and work are all clean and



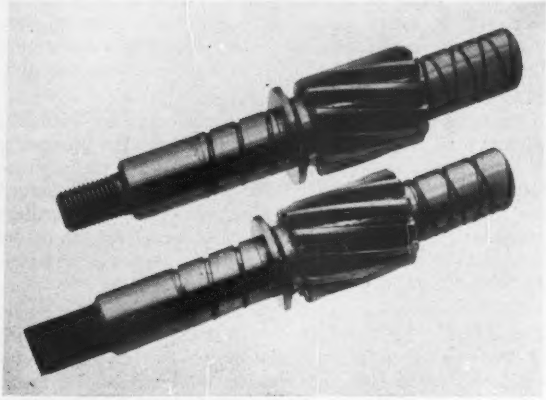
Pre-shaped media in various sections are produced by extruding the mixture in a continuous length and cutting to size before fusing

free from oil or grease. When not in use, steel media, both soft and hardened, should be protected from rusting. For a reasonable period, they can be stored immersed in a concentrated solution of burnishing compound. Steel media, in either soft or hardened condition, are available in a range of five sizes from $\frac{3}{32}$ in to $\frac{3}{8}$ in.

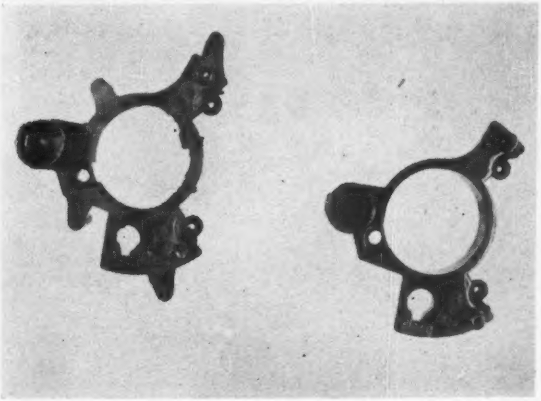
Finishing Compounds

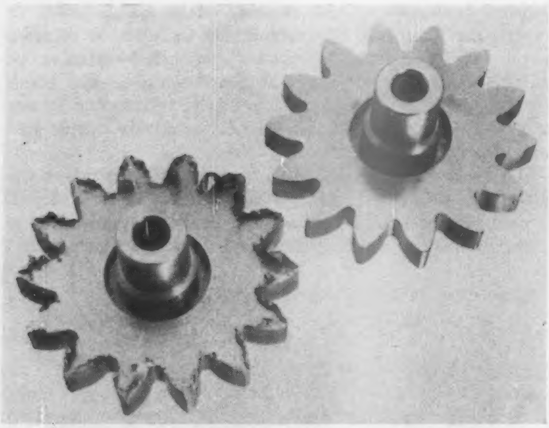
No fewer than thirty standard finishing compounds are supplied. Each is composed of several chemical ingredients for a specific type of finishing process, burnishing, grinding, cleaning, polishing, descaling and combinations of these processes. While some of these compounds may be used as general-purpose compounds for particular metals, it is usually of advantage to select, from the wide range available, a compound specially suited to the work that will produce a superior result. Additionally, the compound selected must perform, in specific instances, such functions as lubrication,

This pinion shaft is finished loose in a barrel in an approximately 2 hr run. The screwed end is protected by a rubber shroud

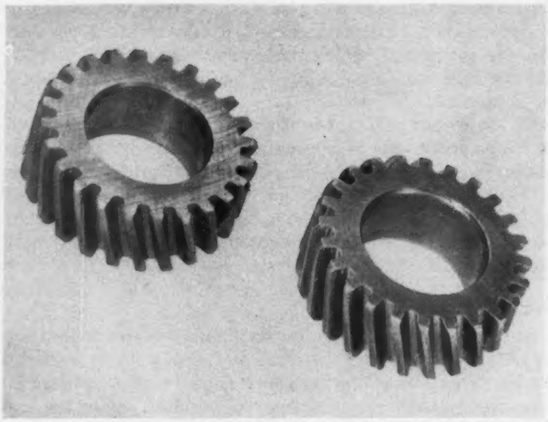


Six hundred of these aluminium die-castings are deflashed, edge-radiused, and polished in a single operation on a DB-400 machine in 2·25 hr





If heavily burred gears are given a preliminary wire-brushing, 45-60 sec, barrelling time is much reduced. On DB-200 machine, 200 parts, 3½ in diameter, 1·5 hr



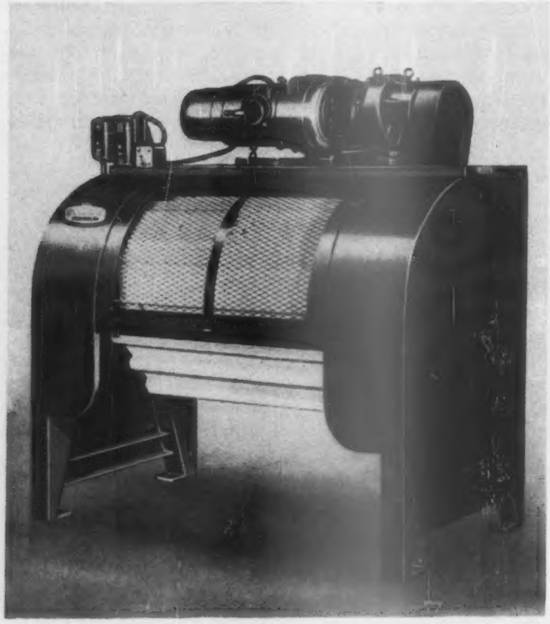
Gears of relatively fine pitch are lightly deburred with small oxide chips in a slowly rotating barrel. On DB-200 machine, 150 parts, 4 in diameter 1·75 hr

water softening, detergency, preventing the loading or glazing of abrasive media, controlling alkalinity, saponifying and emulsifying oils or greases, promoting metal colour, or controlling surface finish.

The Almco Supersheen Division maintains a research laboratory, equipped with a full range of Almco barrelling machines, and will freely advise on the selection of media and compounds for finishing specific components and will process a production load of parts to demonstrate the suitability of the selection and the processing routine. At present this facility is centralized, but eventually fully staffed and equipped laboratories will be maintained at various industrial centres for the greater convenience of users of barrelling equipment, who are thus relieved of the need for costly and time-wasting trial-and-error methods.

All compounds are made in Britain by The Almco Supersheen Division, under the strictest control to ensure consistency. Daily the production of the various compounds is checked and recorded against the quantities of the store chemicals. Any discrepance, indicating incorrect composition of a particular compound, leads to the immediate destruction of the batch concerned. Used on the standard barrelling machines the compounds are expendable; it is more economical to run the water and compound to waste after processing one charge than to expend time in recovering and making additions to restore quality and potency. In the submerged barrelling process, as employed in automatic transfer installations, however, the solutions remain in the tanks at controlled temperatures and do not break down in extended periods of re-use. This results in substantial savings in the cost of materials, which can be used to offset the capital cost of mechanized equipment.

The DB-400 model is typical of the larger machines. Barrel dimensions are 30 in diameter and 48 in long, giving a total capacity of 20·8 ft³. Speed is variable from 6 to 30 rev/min



Processing variables

To specify the finishing process for a particular component, there are a number of variable factors to be considered and a wide range of materials from which to select. There are four grades of abrasive media in various different sizes from 2 in × 1¼ in to $\frac{5}{32}$ in × $\frac{1}{16}$ in; pre-shaped abrasive chips in three different sizes; steel balls, steel shapes, and steel diagonals in five sizes; and thirty different compounds. The number of parts to be treated in each barrel charge must be determined; together with the quantity of media and the volume of water to be used. Also to be decided is the rotational speed of the barrel; the total barrelling time; whether the barrel is unloaded, or drained and flushed out with water ready for a follow-on operation. It will also be necessary, in many instances, to select the most suitable method of drying the finished parts, whether by a dewatering fluid, tumbling in sawdust or a similar absorbent medium, centrifuging, or oven heat.

Printed process specification control cards are available, on which all relevant data can be added for the processing of a particular component. A master control card can be filed and copies can be issued with the job card. It is claimed that this is the first complete system of control for barrelling processes. Owing to the rigid control of media and compound specifications, a satisfactory process is immediately reproducible from the specification card.

With so many variable factors influencing the final result, and as knowledge of the process is mainly empirical, a study of the part to be treated is necessary, or at least highly desirable, before detailed recommendations can be made.

At this stage, the experience of the Company's laboratory staff is most useful and consultation with them and test processing can save the user time and expense.

Multi-process cycles

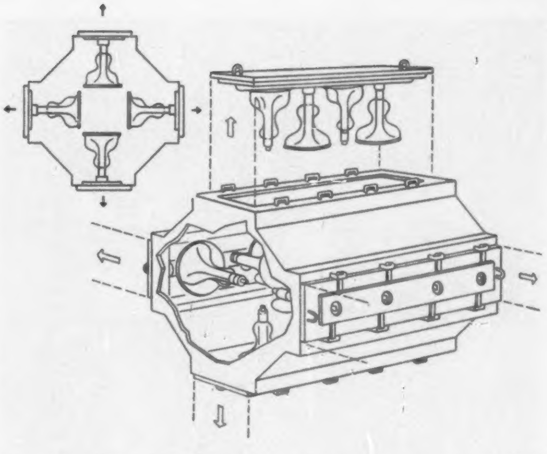
For finishing castings, forgings, heavy stampings, rough machined parts, and other items having rough surfaces, it is usual to apply more than one process cycle. The treatment may comprise grinding with a tough abrasive compound, followed by a run with a polishing, non-abrasive compound. To impart lustre and to improve the surface of parts weighing not more than 6 oz, a burnishing cycle, using steel balls or slugs and a suitable compound, may then be applied. Finishes equal to hand polishing are possible in many instances. It is generally recognized that finishing to such a high standard is not practical on parts exceeding 6 oz in weight. In such cases, the normal method is to perform the major work by barrelling and to complete by a light hand polishing.

Good quality die castings of zinc or aluminium, or brass or steel pressings having good surface conditions, are processed with either a mildly abrasive or a non-abrasive compound, followed by a burnishing treatment for from 30 min to 2 hr duration.

Parts of rectangular shape, having flat surfaces and sharp edges and corners, are difficult to process. Parts having convex surfaces and irregular patterns, particularly those of simple shape and having no holes or recesses, are ideally suited to barrelling. To obtain the best results, chip sizes of $\frac{1}{2}$ in or less should be used. Where this is not practical, on account of holes or recesses, in which the small chips could lodge, larger chip sizes in conjunction with lower barrel speed and a higher solution level will be satisfactory. Alternatively, a mixed chip mass of two or more sizes can be used to good effect.

Chip size

To avoid the lodgement of chips in holes or recesses, a few general rules may be applied. For holes or slots of $\frac{1}{2}$ in or more, a chip size of $\frac{1}{4}$ in larger should be used. Where the chips are required to flow through the hole, the chips should be $\frac{5}{16}$ in smaller than the hole. For holes of less than $\frac{1}{2}$ in diameter, the chips should be from $\frac{3}{16}$ in to $\frac{1}{4}$ in larger to avoid lodgement and $\frac{1}{4}$ in smaller to flow freely through the hole. Should a component have holes or recesses of different sizes that render the selective sizing of chips impractical, the holes may be filled with a wax having a melting point of the order of 140 deg F. After processing, the wax can be removed by hot water or by vapour degreasing. Another practice is to plug holes with rubber inserts. The use of such methods is entirely a question of economics; the cost of individual handling of components may, in some instances, be prohibitive.



Sketch of four-door, multiple-fixture barrel. In this example four components are mounted on each quick-clamping door

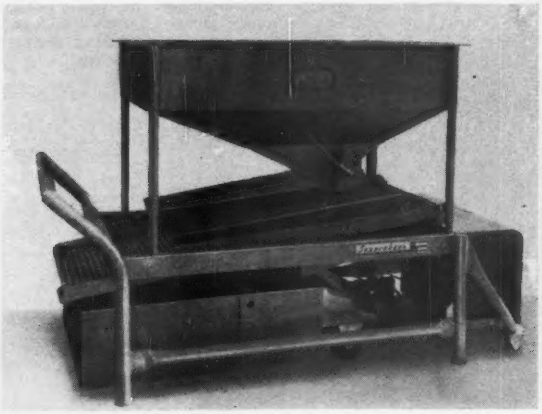
Dome-, cone-, or fan-shaped pressed parts that "nest" with each other are unsuitable for free barrelling, as they rapidly come together during processing and thus large areas escape treatment. The disadvantage can be avoided by fitting small springs or inserts but, again, this method involves individual handling and may be too costly.

On very large components, weighing from 10 lb. to 80 lb, the limitations regarding the geometry of the part encountered in free barrelling do not apply. Such parts are usually jigged in the barrel and do not contact each other. Although fixture barrelling is more expensive than free barrelling, very substantial economies can be effected by the method in comparison with the high cost of hand finishing.

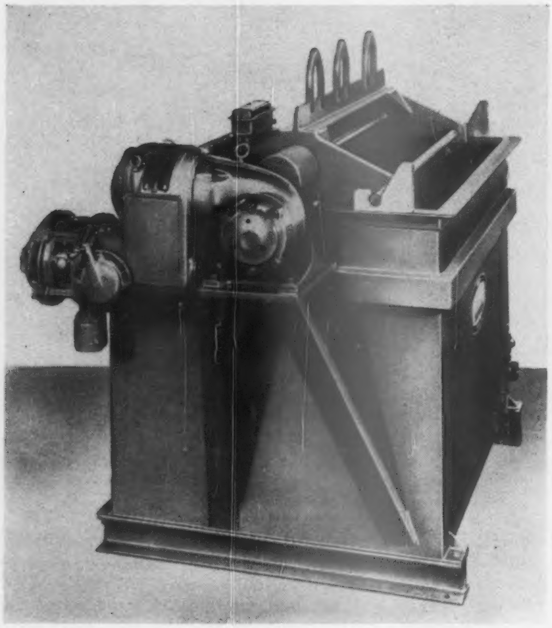
Bright Finishing

For small and medium-sized parts, exceptionally bright finishes are obtainable from chemical- and electro-polishing solutions. However, if normal manufacturing surface defects are to be removed, the cost, both in immersion time and in loss of solution efficiency, becomes prohibitive. Where very high quality finishes are required, the

Portable vibrating screen for separating processed parts from the media. It can be wheeled directly beneath a barrel being unloaded. Screens are interchangeable



Compartmented barrels are used for large components likely to be damaged by part-on-part impingement. Ten parts are treated individually in the machine shown



This "submerged" machine, with independent motor drive for the perforated barrel, forms the basis unit for all looped- or line-type automatic finishing plants

combination of these solutions with barrel finishing presents opportunities for lowering production costs. For example, a low-cost process for polishing aluminium parts has the following sequence:

1. Aluminium etch
2. Wash
3. Acid brightening bath
4. Wash
5. Burnish with steel balls for 45 min to 60 min

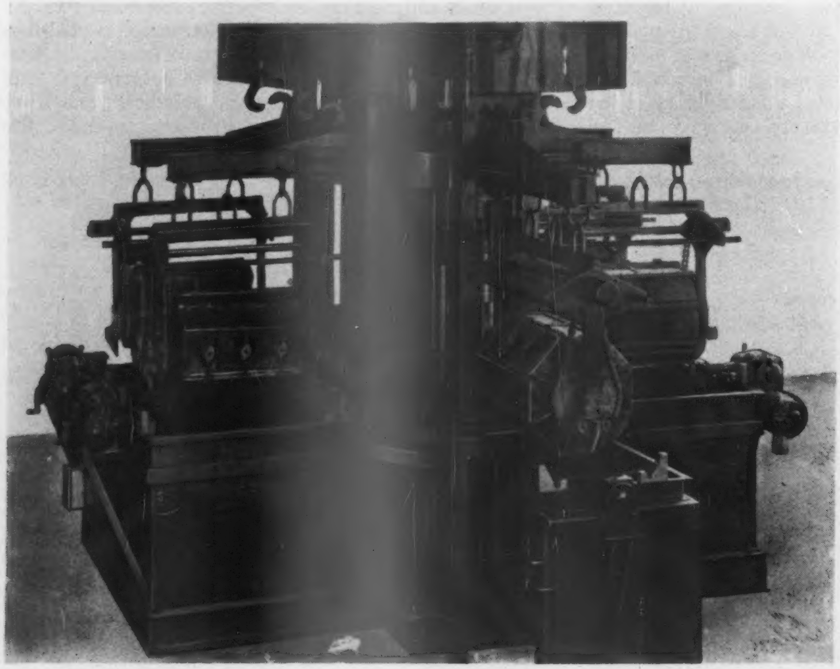
Generally, it is more economical to use large, rather than small, capacity barrels. In many shops, however, the finishing of relatively small batches of different components is the major requirement. Accordingly, the Almco range includes standard machines having barrels of various capacities from 1 ft^3 to 25·8 ft^3 to meet all needs. In all cases, the barrels are octagonal and are lined with rubber or Neoprene, which is preferable to natural rubber since it is unaffected by oil or grease. The Almco organization operates a barrel replacement service, by which a relined barrel is supplied to the user, who then makes the change-over, with the minimum down-time, and returns the worn barrel.

Standard barrelling machines

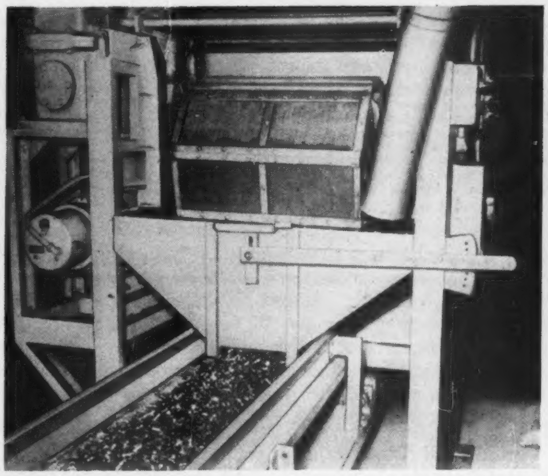
Detachable, watertight, loading and unloading doors are fitted with a cam-type latching arrangement secured by a quarter-turn of a wrench. When follow-on operations are specified, the standard door can be replaced by a perforated draining door, so that water and compound can be changed without discharging the working load of media and parts between operations. Each barrel is equipped with a screw type, release valve, which can bleed off in less than 30 sec any pressure that may be built up in the barrel during long grinding cycles.

The two smaller machines are both twin-barrel, self-contained, fully enclosed models intended for finishing small and medium batches of small parts and for sample processing and development work. They are equipped with a drawer below each barrel, which can be used for loading, unloading, draining or, when fitted with a suitable screen, for separating parts from media. Barrel capacities are 1·0 ft^3 on the smaller model and 2·6 ft^3 on the larger machine and barrel speeds are 10 to 50 rev/min and 15 to 45 rev/min respectively.

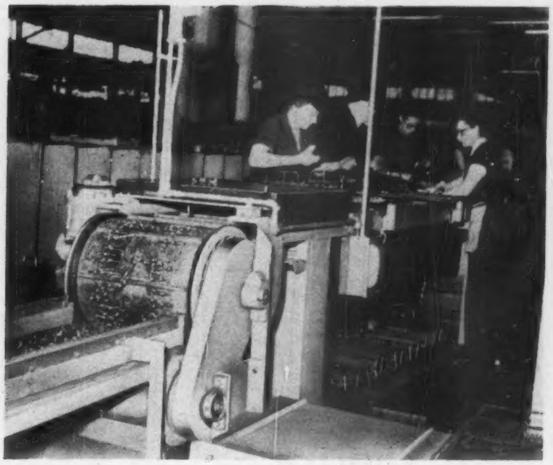
Larger standard machines are all of the single-barrel type and the smallest of these is the DB-50 model illustrated. This has a barrel 22 in diameter and 24 in long, giving a capacity of 5·6 ft^3 in either one or two compartments. It is useful for fairly large parts and can handle a maximum load of parts and media of 800 lb. Other machines of greater capacity are the DB-200, DB-400, and DB-800 models with barrels of 30 in diameter, differing in length at 32 in, 48 in and 60 in respectively. A positive magnetic brake and



On a closed-loop, automatic transfer "skip-station" system processing programmes are changed to suit different batches of work by switching out stations not required at the control panel. Barrels are indexed simultaneously but dwell overhead at idle stations



In unloading the barrel on to the conveyor belt the flow of chips and components is controlled by an end gate to obviate part-on-part impingement



Finished parts are lifted off the unloading conveyor by a magnetic drum and the chips fall to a bucket elevator which raises them to an overhead rotary screen

a forward and reverse switch facilitate positioning of the barrel for loading and unloading. Rotational speed in all cases is from 6 to 30 rev/min. The B-400 machine illustrated is driven by a 5 h.p. motor and has a capacity of 20·8 ft^3.

Fixture barrel machines

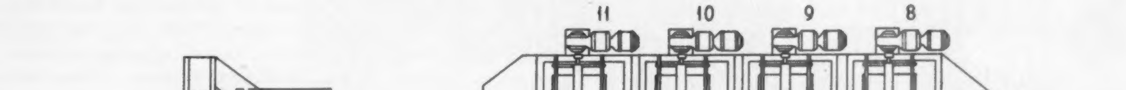
Fixture type barrelling machines are open at the front and the top to facilitate loading and unloading of the fixture doors, by means of an overhead hoist. Fixture barrels can be provided with a plurality of doors (the sketch shows a four-door barrel), which confer several advantages. Substantial savings in compounds are possible. In a typical burnishing operation, where the time cycle is short and only one compound is required, the compound can be used for several loads of parts, since the media does not have to be dumped as in single-door barrel operation. A time cycle is established, an automatic timer is set to stop the machine at regular intervals within the cycle, and at each interval a door with finished parts is taken off. It is immediately replaced by another door loaded with new parts and thus a semi-continuous operation can be maintained to match production requirements. Such machines are available with barrels of from 30 in to 58 in diameter and, in addition to automatic timing, automatic reversing of rotation can be arranged to suit processing specifications.

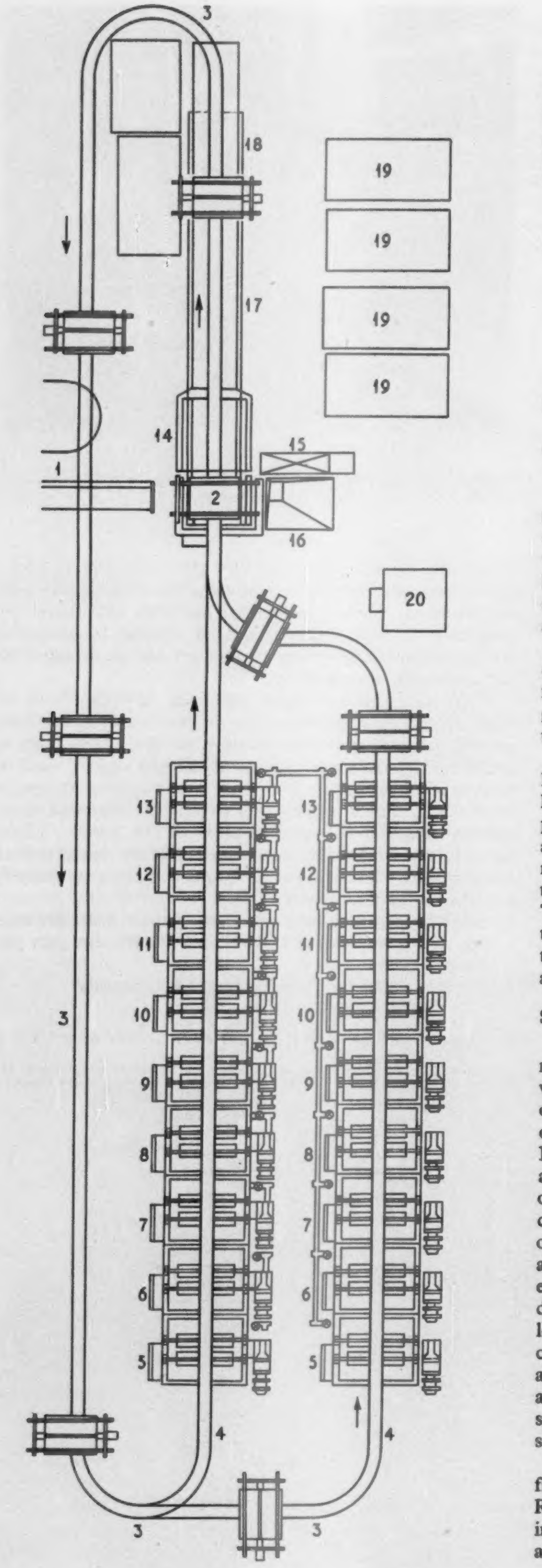
Multi-compartment barrel machines, specially built to meet specific requirements, are of two main types. One pattern has a single door to serve all the compartments, which contain the same compound solution and are used to process identical parts. The other has individual compartment doors, usually angled over two barrel faces and alternatively arranged on opposite sides of the barrel. These are used for short production runs or for the simultaneous processing of different parts. Multi-compartment barrels are effective in cases where:

1. Small quantities of large and heavy parts require processing. They are usually treated individually, one part per compartment
2. Fixtures would be too complicated or too costly

Layout of 9-tank "skip-station" system for finishing centreless-ground parts. Cycle time per barrel 40·5 min, one barrel finished every 4·5 min.

1 load-unload station; 2 control panel; 3 clean and degrease tank; 4, 5, 6, 7, 8, 9 grinding and burnishing tanks; 10 clear water rinse tank; 11 rust-inhibiting tank; 12 parts and media unloaded on to belt conveyor; 13 rotating magnetic separator picks off parts; 14 tunnel drier; 15 pick-off table; 16 bucket elevator; 17 rotary screen classifier; 18 media storage and feed hopper





1 unprocessed parts brought in on belt conveyor; **2** *load-unload station;* **3** *overhead conveyor carrying loaded barrels to finishing lines;* **4** *barrels switched to duplicated finishing lines;* **5** *clear water rinse tanks;* **6, 7, 8, 9, 10** *descaling and grinding tanks;* **11** *neutralizing and grinding tanks;* **12** *burnishing tanks;* **13** *colouring and rust-inhibiting tanks;* **14** *parts and media unloaded on to Neoprene-lined vibrating screen;* **15** *bucket elevator lifts media to rotary classifier;* **16** *media storage and feed hopper;* **17** *tank conveyor for parts, with additional rust-inhibiting solution;* **18** *pick-off and racking table for finished parts;* **19** *parts storage racks;* **20** *finishing compound storage*

Layout of two 9-tank lines for finishing clutch plates. Cycle time per barrel 135 min, one barrel finished every 7·5 min

3. Different parts are to be treated in different compound solutions and with different media
4. It is desirable to eliminate loading and unloading of media after each run.

Semi-automatic handling and separating equipment is available in sizes suitable for all machines. Chuted, self-dumping pans mounted on ball bearing castors can be wheeled under a barrel for unloading or raised by an electric hoist on a tubular structure, also mounted on ball bearing castors, for loading. Screening units are similarly transportable and can be fitted with screens of different mesh from $\frac{1}{16}$ in to $2\frac{1}{2}$ in. The typical mechanical screen illustrated is driven by a self-contained motor of $\frac{1}{2}$ h.p. and the screen frame is vibrated from both sides by an eccentric mechanism giving an elliptical motion, adjustable to length, at rates up to 380 cycles/min.

Power-driven magnetic separators are either permanently located or transportable self-contained units for positively lifting ferrous parts from the abrasive media. A vibrating pan feeder takes mixed parts and media from the hopper and spreads them in a uniform layer, from which the magnetic drum extracts the parts. Parts are then conveyed through a demagnetizer and are collected in a container. The non-magnetic medium is discharged to another container for storage or re-use. Passage of the parts through the demagnetizer eliminates the possiblity of undesirable abrasion from metallic fines and stony particles.

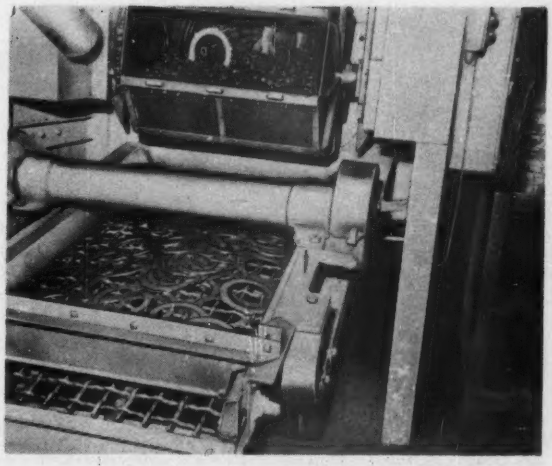
Submerged barrelling

The technique of submerged barrelling was developed to make finishing virtually a continuous process in conformity with modern high-volume production methods. Instead of loading a metal barrel with parts, abrasive media, and compound solution, a perforated, non-metallic barrel with Neoprene-lined end plates is charged with parts and media and rotated while immersed in a tank of compound solution, or of a cleaning or rust-inhibiting solution. Open tanks containing the necessary solutions for a multi-cycle finishing operation are arranged in sequence, and on completion of a cycle in one tank, the barrel is lifted out by overhead hoist equipment, suspended for a short period over the tank to drain, and then lowered into the next tank. The barrel load of parts and media stays intact until all the sequenced cycles are concluded, and the handling of parts before and after sequence processing can be mechanized to save time and labour. Variation of sequence to meet different processing specifications is accomplished by merely changing the solutions.

Submerged barrels are each carried in arms depending from a suspension frame, which seats on the top of the tank. Rotation of the barrel is by a heavy roller chain from a shaft in the suspension frame. Alternative methods of drive are available. Where the number of tanks in an installation

exceeds the number of barrels, a 1·0 h.p. motor, complete with a variable-ratio drive, is mounted on the suspension frame. Under converse conditions, the drive gear with a 1·5 h.p. motor is mounted on the side of the tanks. When the barrel is lowered into the tank, rollers on the suspension frame engage in guide pillars on the tank and the drive is picked up by a pair of coarse-pitched spur pinions. In either case, a limit switch is operated when the frame seats on the tank to start the motor automatically.

In addition to saving on handling time by avoiding unloading and loading between cycles, substantial economies can be effected by the reduction of compounds required, as the solutions are used repeatedly. Recorded cases show reductions of from 80 to 90 per cent in the quantity of compounds used when a change-over from standard batching machines to submerged barrelling machines is made. Maintenance of the solutions at appropriate temperatures ensures their effectiveness for maximum periods of usage. Accordingly, tanks are fitted individually with automatically controlled electric or steam heater units.



Load-unload station on the 18-tank line. Barrels are discharged on to a Neoprene-lined vibrating screen through which the chips fall

Automatic transfer processing

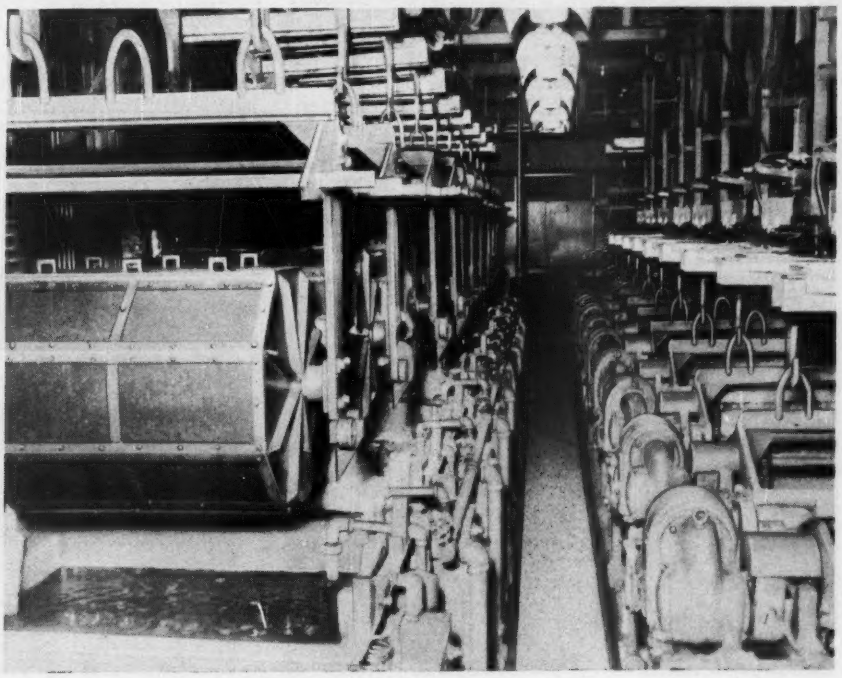
Submerged barrelling presents the possibility of automatic transfer from tank to tank on a timed cycle, after the manner of modern automatic plating systems. The need for such systems to finish components produced at high rates by flow-line methods was quickly recognized. Almco installations are fully developed and in operation in a number of American motor component manufacturing plants. Broadly, they fall into two main types. One, termed the skip-station system, is laid out in a closed loop from a common load-unload station, whereas the other, designated the in-line system, has the tanks arranged in a line terminating with the load-unload station and served by a parallel feeder line.

A six-station, skip system is illustrated and a layout of a ten-station installation is shown in the diagram. As standard, all tanks carry the barrel-driving gear. The barrels are suspended from cantilevered beams, which are raised and lowered by either pneumatic or hydraulic pressure cylinders. Transfer and indexing gear is arranged in the top housing and a latching gear is provided at each indexing station, so that a barrel can be held in the raised position and "skip" a tank not required in a particular process sequence.

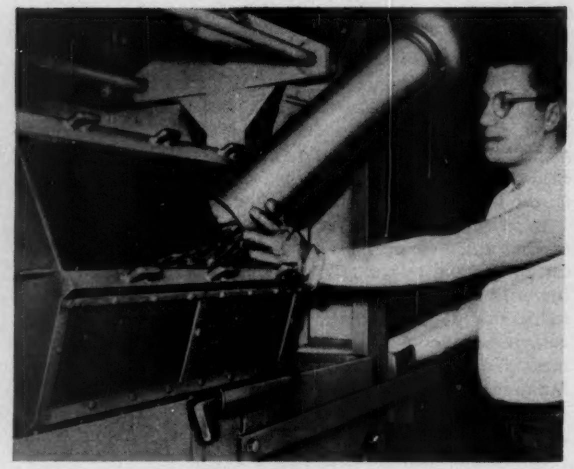
The ten-station installation was supplied to finish machined and centreless-ground steel pistons, plungers and valve spools for vehicle automatic transmission assemblies. Parts must be completely free of burrs, have a minimum edge radius of 0·0005 in, and have a low micro-inch surface finish. The installation provides for the loading and the unloading of a barrel every $4\frac{1}{2}$ min. Actual processing time in each of nine tanks is 3 min, to which must be added $1\frac{1}{2}$ min for drainage and indexing, giving an overall cycle time of $40\frac{1}{2}$ min.

On completing the processing loop and returning to the load-unload station, parts and media are discharged from the barrel to a hopper and thence, at an adjusted rate, to a belt conveyor. This carries them beneath the drum of the magnetic separator, which picks off the parts and conveys

This two-line plant with a total of eighteen tanks finishes one barrel of stamped clutch plates every 7·5 min. Production rate is 3,200 parts per hour



The chip-handling system can deal with 10,000 lb of media per hour. A bucket elevator feeds the chips to a rotating classifier screen which delivers to separate bins in the storage hopper



Barrels are loaded with screened chips to a specified quantity by means of an air-controlled swivel chute from the overhead storage hopper

them through a tunnel drier to the pick-off table. The abrasive medium is delivered by the conveyor to a container from which it is raised by a bucket elevator to a rotary classifying screen, mounted overhead. Here, the chips are sorted in passage over a screen of different mesh and chips still of acceptable size are dropped in a feed hopper for re-use. An air-operated, swivelling chute from this hopper is used for reloading the barrel with the correct quantity of sized chips, and when the parts have been added, the door is re-affixed and the barrel is ready to commence a new finishing cycle.

Duplicated "in line" installation

Clutch plates for an automatic transmission are deburred, edge-radiused, and surface-finished on the line type installation shown in the layout diagram. The overall cycle time is 2 hr 15 min—15 min in each of nine tanks—and by duplicating the tank lines, one barrel of finished parts is unloaded every $7\frac{1}{2}$ min. Barrels are 22 in diameter $\times$ 28 in long and each is loaded with 400 clutch plates, giving an output rate of 3,200 parts per hour.

Unfinished parts are brought to the load-unload station on a shop conveyor, and the loaded barrels are carried to the remote ends of the tank lines by an intermittently operating, overhead conveyor. The barrels are switched alternatively to the two tank lines, along which they are progressed by walking beams, automatically indexing the barrels over the tanks every 15 min. Details of the cycle sequence are given on the layout diagram, and when this is concluded, the contents of the barrel are discharged on to a Neoprene-lined, vibrating screen, which drops the chips to a container and passes the parts to a conveyor immersed in a tank of rust-inhibiting solution. This conveyor raises the parts to the pick-off table, where operators load them into wire baskets, ready for assembly or for storage. The arrangements for the elevation, classification, and storage of media ready for reloading the barrels are identical with those described for the skip system layout.

Tubeless Tyre Machine

An Automatic Inflation Device by Hymatic Engineering Co. Ltd.

AN interesting development by the Automation Division of The Hymatic Engineering Co. Ltd., Redditch, Worcestershire, is an automatic tubeless tyre inflation machine, which is now used by several leading automobile manufacturers in this country. The machine has been devised to speed up what was hitherto a relatively lengthy operation. It enables an operator to inflate a tubeless tyre to within 1 lb/in^2 of the pressure recommended by the manufacturer in approximately six seconds. Through its use, wheels with inflated tyres can be supplied to the assembly line as quickly as they are required and with a great saving in man-hours. Unless a machine of this type is used, each wheel and tyre must be held by a tourniquet, or some other mechanical means of keeping the tyre on the rim, before air can be passed through the valve.

Essentially, the Hymatic unit consists of a steel frame with upper and lower platens situated in the centre. The wheel and the tyre are inserted between the platens. A rubber sealing ring is embedded in the lower platen, while the upper one is hollow so that air can be passed through it into the tyre. A major problem was to design the machine in a manner that would make it suitable for inflating tyres of various sizes. Similar machines are used in the U.S.A., but in that country each manufacturer uses one wheel size only, and the design is comparatively simple. The wide rubber sealing ring on the lower platen of the Hymatic machine enables the unit to be used with a variety of sizes.

Operation is extremely simple. With the tyre loose on the rim, the wheel is placed on the lower platen, so that the underside rim edge is sealed by the rubber ring. The operator then presses the start button; thereafter, the complete sequence is fully automatic. Air is admitted to an accumulator situated at the side. This causes oil on the other side of the accumulator piston to be displaced through a stop valve into the raising cylinder in the machine column beneath the lower platen, and the platen rises, thus sealing the upper tyre wall against the rounded edge of the top platen. At the top of the stroke a sequence valve operates a small ram to close the oil stop valve, and at the same time it opens the air admission valve to the top platen.

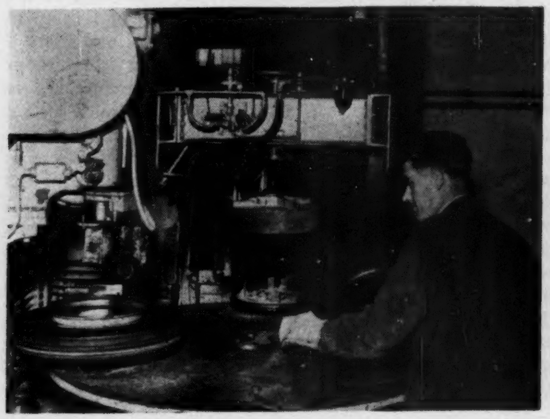
Air flows through and fills the top platen and the tyre until a pre-set pressure, higher than the ultimate pressure for the tyre, is reached. At this point a second sequence valve closes the admission valve and opens the oil stop valve. Release of the oil column allows the lower platen to fall rapidly, producing an "explosion" of the tyre into the wheel, where it forms a perfectly tight seal against the rim. The air expansion in this explosion reduces the tyre pressure to the desired figure. The operator then removes the wheel and places a new wheel and cover on the lower platen.

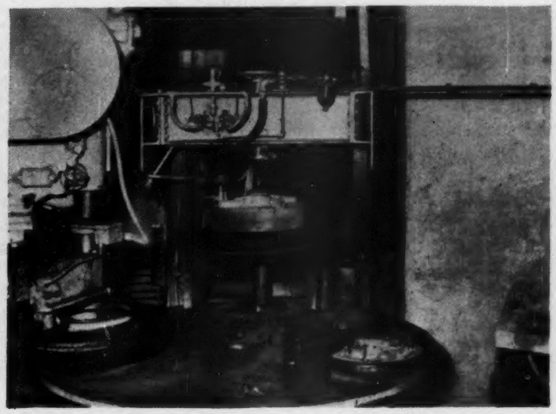
Feeding the machine is facilitated by its open design. Roller conveyors can be arranged to carry the uninflated tyre and its associated wheel to within a few inches of the platens, so that the operator merely pushes the wheel on to the machine ready for inflation and pushes it away to a conveyor on the other side after inflation. When required, the machine can be supplied with all the hydraulic and compressed air components duplicated. In the event of a failure, the operator merely switches over to the standby system while repairs are carried out.

A further development of this machine has recently been put into operation at the Luton works of Vauxhall Motors Ltd. With this machine, which is shown in the accompanying illustrations, the inflation device forms part of a three-station circular transfer machine arranged to fit and inflate the tyres. The machine is some 15 ft in diameter, and the compressed air supply, which is needed in any case to inflate the tyres, is also used for the feed and transfer motions.

At the first station a wheel and tyre are loaded manually, the tyre being partly placed over the wheel rim. Operation of a press button then starts the automatic cycle. The table indexes and the wheel and tyre are carried to the second station where a wheel roller pushes the tyre on to the wheel. At the third station the lower platen, and with it the wheel and tyre, is raised until the tyre makes contact with the upper platen in the inflation position. On the completion of inflation, the lower platen descends, the tyre explodes into the wheel and the correctly inflated tyre and wheel assembly is automatically discharged on to a conveyor for transfer to the assembly line. On this machine the air valves are controlled electrically by solenoids, and correct sequencing is effected by a series of limit switches.

Automatic inflation machine at Vauxhall Motors. At the left, the operator is loading a wheel and tyre; the illustration at the right shows a tyre being rolled into the wheel at the second station and a tyre being inflated at the third station





EXHAUST BRAKE PERFORMANCE

Results of Tests by the Road Research Laboratory on a Commercial Vehicle

R. N. KEMP, A.F.Ae.S.*

THIS article describes the results of tests made on the braking performance of a commercial vehicle fitted with an exhaust brake, so arranged as to operate before the normal wheel brakes. It was found that a saving of about 20 per cent in usage of the main wheel brakes could be expected in normal traffic; on gradients a saving of about 33 per cent might be obtained. Some notes on auxiliary brakes for commercial and public service vehicles are appended.

Auxiliary brakes have been in use on heavy lorries, buses and coaches in the hilly and mountainous districts of Europe for many years to reduce the work required from the main wheel brakes during long descents, so reducing the possibilities of overheating and resultant brake fade. In Great Britain, with its comparatively few long steep gradients, it is only recently that the use of an auxiliary brake has been considered desirable. With the tendency towards increased gross vehicle weights and with the raising of the speed limit from 20 m.p.h. to 30 m.p.h. for heavy commercial vehicles, auxiliary brakes will probably become more common. The simplest form of auxiliary brake and one that can readily be fitted to many commercial and public service vehicles is the exhaust brake. This article describes some tests made with a particular commercial vehicle fitted with an exhaust brake exhaust brake, the degree of braking used being continuously recorded during each descent. All tests were made in top gear from an initial speed of about 20 m.p.h. and the speed of the vehicle was kept to between 20 and 25 m.p.h. speedometer reading during the whole descent. Runs in which baulking occurred due to other traffic were not included. The results are given in Table III.

Description of tests

An ex-military four-wheeled tractor with a 7·7-litre compression-ignition engine, laden to a gross weight of 10 tons 18 cwt, was fitted with an electrically-operated exhaust brake. The main brakes of this vehicle were of the compressed-air type and were in good condition. The exhaust brake was operated by a switch on the brake pedal, care being taken to ensure that there was sufficient difference in "feel" for the driver to be able to apply the exhaust brake without the main brake. A pressure gauge fitted to the exhaust manifold showed that the maximum pressure with the exhaust brake in use was 37 lb/in² and all tests at 20 m.p.h. or more were made at this pressure. Below this speed in top gear the exhaust back pressure fell off. The main brakes were operated at a pressure of 90 lb/in².

Measurement of braking performance. A fifth wheel [1] was used to measure the minimum braking distance obtainable using (a) the main wheel brakes, (b) the main wheel brakes and exhaust brake, (c) the exhaust brake and engine only and (d) the engine only. In (a) and (b) the vehicle was brought to a stop but in (c) and (d) the distance measured was from the point of application of the brakes to the point at which the engine running at "tick over" speed tended to propel the vehicle again (about 5 m.p.h.). The time and distance required to reduce speed from 30 to 20 m.p.h. and from 20 to 10 m.p.h. as in (c) and (d) were also measured, the speed and distance being measured by the fifth wheel and the time by stop watch.

All the tests were made on an aerodrome runway or test track, the surfaces of both being reasonably level and in good condition. All tests were made in top gear with the same vehicle, driver and load. Six runs were made in each case. The results are given in Tables I and II.

Effect of exhaust brake in descending hills. Three hills of different gradients, Fig. 1, were selected for these tests and six descents were made on each hill with and without the exhaust brake, the degree of braking used being continuously recorded during each descent. (see above)

Method of comparing the work done by the wheel brakes. To assess the saving in wear of the main wheel brakes when descending hills, without making the large number of test runs necessary to produce measurable wear of the brake lining, it was assumed that for a given vehicle maintained at a steady speed the wear would be proportional to the magnitude and duration of the decelerations used in maintaining that speed. As the vehicle tested was fitted with a compressed-air braking system in which the distance the brake pedal was depressed was proportional to the force exerted on the brakes and, hence, provided the wheels did not lock, to the deceleration achieved, the pedal movement and duration of each stage of movement were recorded. This was done by arranging for the brake pedal to move over a series of seven contacts, each contact operating one pen of a continuous paper recorder. A further pen gave a time scale so that, from the record, the length of time the brake pedal had been depressed by the various steps of movement could be evaluated. Fig. 2 shows a specimen record. A separate calibration of pedal movement against Tapley meter readings, Fig. 3, gave the range of deceleration covered by each pen. The work done by the main wheel brakes under different conditions was evaluated by comparing the summation, for each condition, of the individual steps in deceleration (represented by brake pedal movement) multiplied by the time for which each deceleration acted. At constant speed this is proportional to the work done in braking. A typical set of observations on one hill is shown in Fig. 4 and further results in Figs. 5 and 6.

Discussion of results

As shown in Table I and Fig. 5, the increase in the maximum braking performance of the test vehicle obtained by fitting an exhaust brake was slight, the minimum braking distance being reduced by 1 ft (2 per cent) at 20 m.p.h. and by 6 ft (6 per cent) at 30 m.p.h. However, it is clear from the results that there is a considerable saving in the work required from the main brakes, the actual amount depending on the gradient, ranging from 33 per cent for a hill of 1 in 10 to 100 per cent for a slope of 1 in 22 (Table III and Fig. 6). A greater saving could have been obtained on the steeper gradients by using a lower gear but no measurements have been made. If the exhaust brake control is arranged so that it must be applied before the main wheel brakes are used, a further saving will be effected during brake applications in normal traffic conditions. No figures are available for the decelerations used by commercial vehicles in normal circumstances but it is unlikely that they will be higher than those for cars, the mean of which has been found in previous work by the Laboratory to be about 0·15 *g*. If therefore the mean deceleration obtained with an exhaust brake is 0·03 *g* (from Table I, the lowest recorded during these tests), the exhaust brake, of course, being applied each time and for the same duration as the main brakes, the saving in effort required from the main wheel brakes would be 20 per cent. From the

*Road Research Laboratory, Department of Scientific and Industrial Research.

mean decelerations achieved using the exhaust brake alone to slow the vehicle, as given in Table II, an even greater saving could be expected. The total reduction in the use of the main wheel brakes would, of course, depend on the proportion and severity of gradients on a particular route, though it is clear that considerable saving in brake usage would occur on all routes and that on the less hilly routes the greatest saving might be achieved, since the deceleration produced by the exhaust brake would constitute a bigger proportion of the decelerations employed on each occasion.

It will be seen from Table II that the mean deceleration using both engine alone and engine with exhaust brake to reduce speed from 30 to 20 m.p.h. was higher than that for reducing speed from 20 to 10 m.p.h. If, as shown by Johannis in laboratory tests[2], the efficiency of the exhaust brake was higher at greater engine speeds, it would be expected that the increase, due to a higher speed, would be more with the exhaust brake in operation. Table II shows, however, that for vehicle A the increase was the same as when using the engine alone and for vehicle C it was less. This indicates that for these vehicles any increase in performance of the exhaust brake due to greater engine speed is negligible within this speed range. As these tests were on different vehicles under different conditions, the variation of the results in Table II is not unexpected.

Conclusions

The main findings were as follows:—

(1) There was only a small increase in the maximum braking performance of the vehicle when using the exhaust brake in addition to the normal wheel brakes for an emergency stop.

(2) The mean overall deceleration of the vehicle when using the engine alone to retard it was $0{\cdot}015\,g$; with the exhaust brake in operation, the mean overall deceleration increased to $0{\cdot}031\,g$.

(3) To maintain a steady speed of about 20 m.p.h. on a down gradient with the vehicle in top gear, it was not necessary to use the main brakes on slopes of 1 in 22 or less. On gradients of 1 in 10, a saving of about 33 per cent in usage of the main brake was obtained.

(4) In normal traffic applications a saving of about 20 per cent in usage of the main wheel brakes would have been expected.

The work described in this article was carried out as part of the programme of the Road Research Board of the Department of Scientific and Industrial Research. The article is published by permission of the Director of Road Research.

TABLE I. BRAKING PERFORMANCE ON A LEVEL SURFACE OF A VEHICLE FITTED WITH AN EXHAUST BRAKE

Method of braking	Speed (m.p.h.)	Minimum braking distance (ft) (mean of 6 stops)	Corresponding mean deceleration (per cent g)	Standard deviation of braking distance (ft)
Main brakes	20	51	26	1·34
	30	101	30	3·16
Main brakes plus exhaust brake	20	50	27	1·61
	30	95	31	1·34
Exhaust brake only*	20	416	3·0	15·8
	30	919	3·2	19·1
Engine only*	20	924	1·4	26·5
	30	1712	1·7	19·2

**Vehicle not brought to a stop but measurements made to point at which load on engine reversed, that is, about 5 m.p.h.*

Appendix

SOME NOTES ON AUXILIARY BRAKES FOR COMMERCIAL AND PUBLIC SERVICE VEHICLES

Types of auxiliary brake

Auxiliary brakes may be divided into two classes, namely transmission brakes, which work on the vehicle drive on the output side of the gear box, and engine brakes. The basic differences between these two classes are that while the transmission brake, when applied, cannot be disconnected from the driven wheels it is possible for the engine brake to be disconnected by selecting neutral gear or disengaging the clutch. An advantage of the engine brake, however, is that the speed of the engine, and therefore the total horse-power dissipated by the brakes, can be increased (by changing to a low gear) without increasing the speed of the vehicle. In other types of auxiliary brake, the energy dissipation by the brake is directly proportional to the speed of the vehicle.

Transmission brakes

Transmission brakes of the drum or disc type are used in many Continental and American vehicles as parking brakes or to relieve the work of the main wheel brakes. These

TABLE II. PERFORMANCE OF VEHICLE ENGINE IN REDUCING SPEED BY 10 m.p.h. WITH AND WITHOUT EXHAUST BRAKE IN OPERATION

Gross vehicle weight and engine capacity	Method of braking	30 m.p.h.—20 m.p.h. Time (s)	Mean deceleration (per cent g)	Per cent increase in deceleration with exhaust brake	20 m.p.h.—10 m.p.h. Time (s)	Mean deceleration (per cent g)	Per cent increase in deceleration with exhaust brake	Increase in deceleration in higher speed range (per cent g)
A. 10·9 tons	Exhaust brake	8·9	5·1	46	10·8	4·3	59	0·8
7·7 litre	Engine only	12·9	3·5		17·2	2·7		0·8
B. 9 tons	Exhaust brake	7·0	6·5	124				
7·4 litre	Engine only	16·0	2·9					
C. 7·6 tons	Exhaust brake	5·0	9·2	70	5·5	8·3	98	0·9
4·73 litre	Engine only	8·5	5·4		11·0	4·2		1·2

Vehicle A; Road Research Laboratory tests (mean of 6 tests). Vehicles B and C; published data (see references 14, 15)

EXHAUST BRAKE PERFORMANCE

Results of Tests by the Road Research Laboratory on a Commercial Vehicle

R. N. Kemp, A.F.Ae.S.*

THIS article describes the results of tests made on the braking performance of a commercial vehicle fitted with an exhaust brake, so arranged as to operate before the normal wheel brakes. It was found that a saving of about 20 per cent in usage of the main wheel brakes could be expected in normal traffic; on gradients a saving of about 33 per cent might be obtained. Some notes on auxiliary brakes for commercial and public service vehicles are appended.

Auxiliary brakes have been in use on heavy lorries, buses and coaches in the hilly and mountainous districts of Europe for many years to reduce the work required from the main wheel brakes during long descents, so reducing the possibilities of overheating and resultant brake fade. In Great Britain, with its comparatively few long steep gradients, it is only recently that the use of an auxiliary brake has been considered desirable. With the tendency towards increased gross vehicle weights and with the raising of the speed limit from 20 m.p.h. to 30 m.p.h. for heavy commercial vehicles, auxiliary brakes will probably become more common. The simplest form of auxiliary brake and one that can readily be fitted to many commercial and public service vehicles is the exhaust brake. This article describes some tests made with a particular commercial vehicle fitted with an exhaust brake.

Description of tests

An ex-military four-wheeled tractor with a 7·7-litre compression-ignition engine, laden to a gross weight of 10 tons 18 cwt, was fitted with an electrically-operated exhaust brake. The main brakes of this vehicle were of the compressed-air type and were in good condition. The exhaust brake was operated by a switch on the brake pedal, care being taken to ensure that there was sufficient difference in "feel" for the driver to be able to apply the exhaust brake without the main brake. A pressure gauge fitted to the exhaust manifold showed that the maximum pressure with the exhaust brake in use was 37 lb/in² and all tests at 20 m.p.h. or more were made at this pressure. Below this speed in top gear the exhaust back pressure fell off. The main brakes were operated at a pressure of 90 lb/in².

Measurement of braking performance. A fifth wheel [1] was used to measure the minimum braking distance obtainable using (a) the main wheel brakes, (b) the main wheel brakes and exhaust brake, (c) the exhaust brake and engine only and (d) the engine only. In (a) and (b) the vehicle was brought to a stop but in (c) and (d) the distance measured was from the point of application of the brakes to the point at which the engine running at "tick over" speed tended to propel the vehicle again (about 5 m.p.h.). The time and distance required to reduce speed from 30 to 20 m.p.h. and from 20 to 10 m.p.h. as in (c) and (d) were also measured, the speed and distance being measured by the fifth wheel and the time by stop watch.

All the tests were made on an aerodrome runway or test track, the surfaces of both being reasonably level and in good condition. All tests were made in top gear with the same vehicle, driver and load. Six runs were made in each case. The results are given in Tables I and II.

Effect of exhaust brake in descending hills. Three hills of different gradients, Fig. 1, were selected for these tests and six descents were made on each hill with and without the exhaust brake, the degree of braking used being continuously recorded during each descent. All tests were made in top gear from an initial speed of about 20 m.p.h. and the speed of the vehicle was kept to between 20 and 25 m.p.h. speedometer reading during the whole descent. Runs in which baulking occurred due to other traffic were not included. The results are given in Table III.

Method of comparing the work done by the wheel brakes. To assess the saving in wear of the main wheel brakes when descending hills, without making the large number of test runs necessary to produce measurable wear of the brake lining, it was assumed that for a given vehicle maintained at a steady speed the wear would be proportional to the magnitude and duration of the decelerations used in maintaining that speed. As the vehicle tested was fitted with a compressed-air braking system in which the distance the brake pedal was depressed was proportional to the force exerted on the brakes and, hence, provided the wheels did not lock, to the deceleration achieved, the pedal movement and duration of each stage of movement were recorded. This was done by arranging for the brake pedal to move over a series of seven contacts, each contact operating one pen of a continuous paper recorder. A further pen gave a time scale so that, from the record, the length of time the brake pedal had been depressed by the various steps of movement could be evaluated. Fig. 2 shows a specimen record. A separate calibration of pedal movement against Tapley meter readings, Fig. 3, gave the range of deceleration covered by each pen. The work done by the main wheel brakes under different conditions was evaluated by comparing the summation, for each condition, of the individual steps in deceleration (represented by brake pedal movement) multiplied by the time for which each deceleration acted. At constant speed this is proportional to the work done in braking. A typical set of observations on one hill is shown in Fig. 4 and further results in Figs. 5 and 6.

Discussion of results

As shown in Table I and Fig. 5, the increase in the maximum braking performance of the test vehicle obtained by fitting an exhaust brake was slight, the minimum braking distance being reduced by 1 ft (2 per cent) at 20 m.p.h. and by 6 ft (6 per cent) at 30 m.p.h. However, it is clear from the results that there is a considerable saving in the work required from the main brakes, the actual amount depending on the gradient, ranging from 33 per cent for a hill of 1 in 10 to 100 per cent for a slope of 1 in 22 (Table III and Fig. 6). A greater saving could have been obtained on the steeper gradients by using a lower gear but no measurements have been made. If the exhaust brake control is arranged so that it must be applied before the main wheel brakes are used, a further saving will be effected during brake applications in normal traffic conditions. No figures are available for the decelerations used by commercial vehicles in normal circumstances but it is unlikely that they will be higher than those for cars, the mean of which has been found in previous work by the Laboratory to be about $0{\cdot}15\,g$. If therefore the mean deceleration obtained with an exhaust brake is $0{\cdot}03\,g$ (from Table I, the lowest recorded during these tests), the exhaust brake, of course, being applied each time and for the same duration as the main brakes, the saving in effort required from the main wheel brakes would be 20 per cent. From the

*Road Research Laboratory, Department of Scientific and Industrial Research.

mean decelerations achieved using the exhaust brake alone to slow the vehicle, as given in Table II, an even greater saving could be expected. The total reduction in the use of the main wheel brakes would, of course, depend on the proportion and severity of gradients on a particular route, though it is clear that considerable saving in brake usage would occur on all routes and that on the less hilly routes the greatest saving might be achieved, since the deceleration produced by the exhaust brake would constitute a bigger proportion of the decelerations employed on each occasion.

It will be seen from Table II that the mean deceleration using both engine alone and engine with exhaust brake to reduce speed from 30 to 20 m.p.h. was higher than that for reducing speed from 20 to 10 m.p.h. If, as shown by Johannis in laboratory tests[2], the efficiency of the exhaust brake was higher at greater engine speeds, it would be expected that the increase, due to a higher speed, would be more with the exhaust brake in operation. Table II shows, however, that for vehicle A the increase was the same as when using the engine alone and for vehicle C it was less. This indicates that for these vehicles any increase in performance of the exhaust brake due to greater engine speed is negligible within this speed range. As these tests were on different vehicles under different conditions, the variation of the results in Table II is not unexpected.

Conclusions

The main findings were as follows:—

(1) There was only a small increase in the maximum braking performance of the vehicle when using the exhaust brake in addition to the normal wheel brakes for an emergency stop.

(2) The mean overall deceleration of the vehicle when using the engine alone to retard it was $0{\cdot}015\,g$; with the exhaust brake in operation, the mean overall deceleration increased to $0{\cdot}031\,g$.

(3) To maintain a steady speed of about 20 m.p.h. on a down gradient with the vehicle in top gear, it was not necessary to use the main brakes on slopes of 1 in 22 or less. On gradients of 1 in 10, a saving of about 33 per cent in usage of the main brake was obtained.

(4) In normal traffic applications a saving of about 20 per cent in usage of the main wheel brakes would have been expected.

The work described in this article was carried out as part of the programme of the Road Research Board of the Department of Scientific and Industrial Research. The article is published by permission of the Director of Road Research.

TABLE I. BRAKING PERFORMANCE ON A LEVEL SURFACE OF A VEHICLE FITTED WITH AN EXHAUST BRAKE

Method of braking	Speed (m.p.h.)	Minimum braking distance (ft) (mean of 6 stops)	Corresponding mean deceleration (per cent g)	Standard deviation of braking distance (ft)
Main brakes	20	51	26	1·34
	30	101	30	3·16
Main brakes plus exhaust brake	20	50	27	1·61
	30	95	31	1·34
Exhaust brake only*	20	416	3·0	15·8
	30	919	3·2	19·1
Engine only*	20	924	1·4	26·5
	30	1712	1·7	19·2

** Vehicle not brought to a stop but measurements made to point at which load on engine reversed, that is, about 5 m.p.h.*

Appendix

SOME NOTES ON AUXILIARY BRAKES FOR COMMERCIAL AND PUBLIC SERVICE VEHICLES

Types of auxiliary brake

Auxiliary brakes may be divided into two classes, namely transmission brakes, which work on the vehicle drive on the output side of the gear box, and engine brakes. The basic differences between these two classes are that while the transmission brake, when applied, cannot be disconnected from the driven wheels it is possible for the engine brake to be disconnected by selecting neutral gear or disengaging the clutch. An advantage of the engine brake, however, is that the speed of the engine, and therefore the total horsepower dissipated by the brakes, can be increased (by changing to a low gear) without increasing the speed of the vehicle. In other types of auxiliary brake, the energy dissipation by the brake is directly proportional to the speed of the vehicle.

Transmission brakes

Transmission brakes of the drum or disc type are used in many Continental and American vehicles as parking brakes or to relieve the work of the main wheel brakes. These

TABLE II. PERFORMANCE OF VEHICLE ENGINE IN REDUCING SPEED BY 10 m.p.h. WITH AND WITHOUT EXHAUST BRAKE IN OPERATION

Gross vehicle weight and engine capacity	Method of braking	30 m.p.h.—20 m.p.h.			20 m.p.h.—10 m.p.h.			Increase in deceleration in higher speed range (per cent g)
		Time (s)	Mean deceleration (per cent g)	Per cent increase in deceleration with exhaust brake	Time (s)	Mean deceleration (per cent g)	Per cent increase in deceleration with exhaust brake	
A. 10·9 tons	Exhaust brake	8·9	5·1	46	10·8	4·3	59	0·8
7·7 litre	Engine only	12·9	3·5		17·2	2·7		0·8
B. 9 tons	Exhaust brake	7·0	6·5	124				
7·4 litre	Engine only	16·0	2·9					
C. 7·6 tons	Exhaust brake	5·0	9·2	70	5·5	8·3	98	0·9
4·73 litre	Engine only	8·5	5·4		11·0	4·2		1·2

Vehicle A; Road Research Laboratory tests (mean of 6 tests). Vehicles B and C; published data (see references 14, 15)

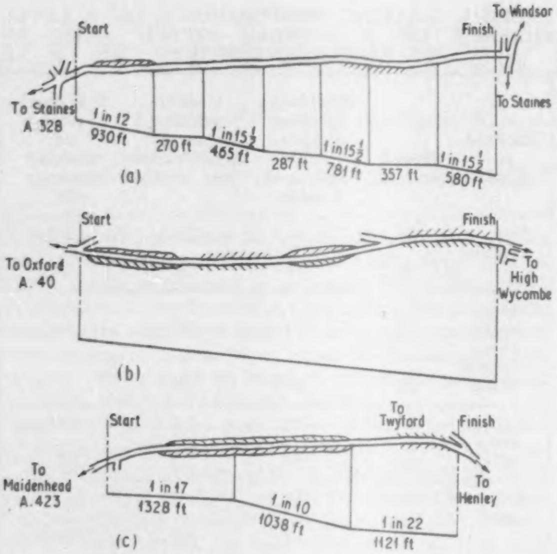
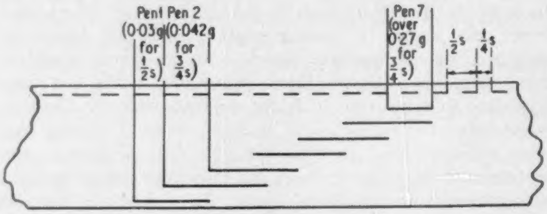


Fig. 1. (left). Plan of the gradients used for the tests
(a) Priest Hill A.328. Length 3,830 ft. Average gradient 1 in 18
(b) Dashwood Hill A.40. Length 5,364 ft. Average gradient 1 in 22
(c) Remenham Hill A.423. Length 3,487 ft. Average gradient 1 in 15

Fig. 2. A specimen record showing maximum application of the footbrake



brakes, being of the friction type, require periodic adjustment and replacement of the friction material. Two types of transmission brake specifically designed as auxiliary brakes are the Telma and Westral brakes.

Telma brake. A disc attached to the propeller shaft of the vehicle revolves between two fixed plates. Electromagnets on these plates face the disc and on being supplied with current from the vehicle battery produce eddy currents so that the resulting magnetic forces cause a retarding or braking moment on the disc and therefore on the vehicle transmission and driven wheels. Tests made in Germany[3] showed that a maximum deceleration of 1·5 metres per second ($0{\cdot}15\,g$) can be obtained and that this maximum is available at low speeds, a desirable feature in an auxiliary brake. Figure 7 compares characteristic curves of braking torque against propeller shaft speed for a normal friction-type brake and for the Telma and Super Telma brakes. The latter works on the same principle as the Telma brake but has been developed for vehicles, such as buses and delivery vans, which do not attain high speeds but which stop frequently. The Telma type of brake is of course liable to failure through normal electrical faults. Cases have also been reported[3] of the disc becoming red hot and heating up the floor of the vehicle.

Westral brake. This is a friction brake of the disc type fitted to the propeller shaft; it is water-cooled by the engine cooling system. It is said to be efficient in operation but does require periodic adjustment. It is comparatively intricate and requires modification to the propeller shaft. Any failure in the cooling system might cause considerable overheating of the friction surfaces with possible loss of efficiency.

Engine brakes

The engine of a vehicle in motion will, if the throttle is closed, exert a retarding force on that vehicle, as part of its kinetic energy will have to overcome the frictional, pumping and other mechanical losses in the engine. This retarding force is, however, very limited and various methods have been devised for increasing the effectiveness of an engine as a brake. There are two main methods of achieving this in four-stroke engines. Both obtain their braking torque in the same way by causing the engine to work as a compressor, the resultant torque being transmitted through the gear box and transmission to the driven wheels as a retarding force. Some work has been done on fluid friction brakes and also on two-stroke diesel engines to improve their braking power, which is less than that of an unbraked four-stroke engine.

TABLE III. SAVING IN BRAKING EFFORT WHEN USING EXHAUST BRAKE ON DIFFERENT GRADIENTS

(Each figure is mean of 6 runs)

Site of test (a)	Mean gradient (b)	Length of test portion (ft) (c)	Mean speed for descent (m.p.h.) (d)	Measure of total braking effort of main wheel brakes (per cent $g \times$ time): Without exhaust brake (e)	With exhaust brake (f)	Percentage saving in braking effort with exhaust brake $\frac{e-f}{e} \times 100$
Priest Hill:						
Mean for whole hill	1 in 18	3830	24	291	84	71
Part of hill	1 in 15½	1246	—	51	19	63
Part of hill	1 in 12	930	—	96	48	50
Dashwood Hill:						
Mean for whole hill	1 in 22	5364	24	301	3	99
Remenham Hill:						
Mean for whole hill	1 in 15	3486	25	345	177	49
Part of hill	1 in 22	1121	—	68	0	100
Part of hill	1 in 10	1038	—	228	153	33
Part of hill	1 in 17	1328	—	87	38	56

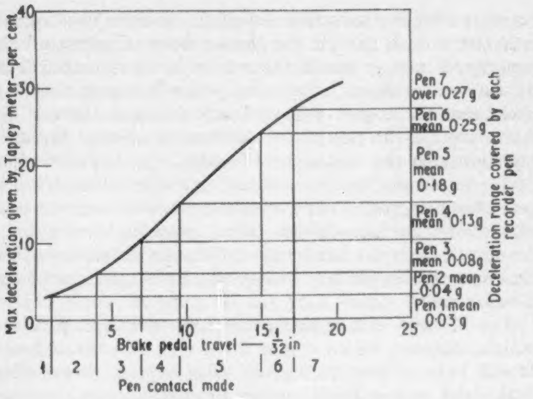
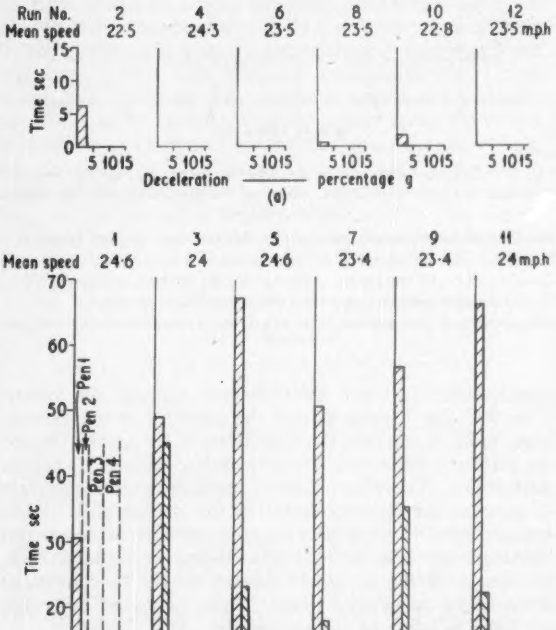


Fig. 3. Calibration curve of brake movement against deceleration (Tapley Meter)

Some success appears to have been achieved with two-stroke engines on the Continent by fitting an exhaust valve to the cylinder head to replace the normal scavenging ports; with at least one type, further braking can be obtained by changing the camshaft timing so that the exhaust valve closes every time the piston moves upwards.[4]

Hydraulic retarders. In this type of brake, tractive resistance is achieved by causing the vehicle to drive an impeller or rotor in a suitable fluid, work done by the impeller being converted into heat in the fluid and finally dissipated through a cooling system. An advantage of this type of retarder is that the retarding force is greater at the higher vehicle speeds.[5]

Double variable camshaft brake. In this type of brake, mechanical alteration of the valve operating cams changes the valve cycle of the engine and causes it to work as a two-stroke compressor. It is not, however, readily adaptable to any vehicle and would normally need to be incorporated in the design of the engine.



Exhaust brake or obturator. This is the type of auxiliary brake which is most extensively used on the Continent and is becoming increasingly popular in Great Britain. It causes the engine to work as a compressor on the exhaust as well as on the normal compression stroke, so increasing the retarding torque of the engine. Comparatively cheap and readily fitted to existing vehicles which have four-stroke engines, the exhaust brake consists of a throttle in the exhaust pipe of the engine which can be closed either by mechanical, electrical or pneumatic means or a combination of these means.

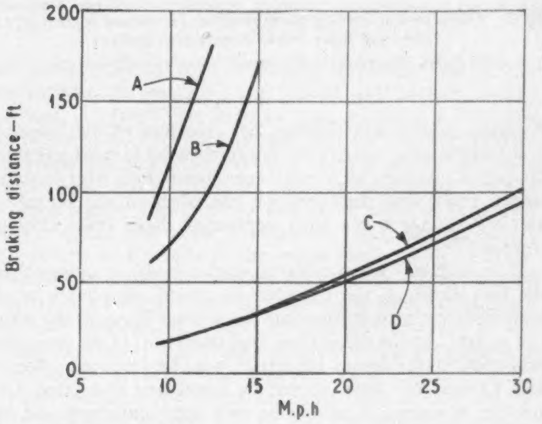


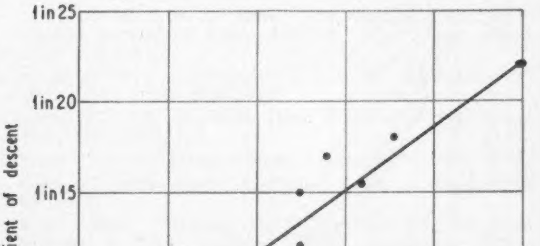
Fig. 5. Minimum braking distances on a level surface from various speeds

(A) *Engine only* **(B)** *Exhaust brake only*
(C) *Main brakes* **(D)** *Main and exhaust brakes*

Fig. 4. (left). Diagram showing the degree and duration of use of main brakes on a 1 in 22 gradient in six descents with and without the use of an exhaust brake

(a) *Usage of main brakes with exhaust brake in operation*
(b) *Usage of main brakes without exhaust brake*

Fig. 6. Percentage reduction in work done by the main brakes through the use of an exhaust brake on different gradients



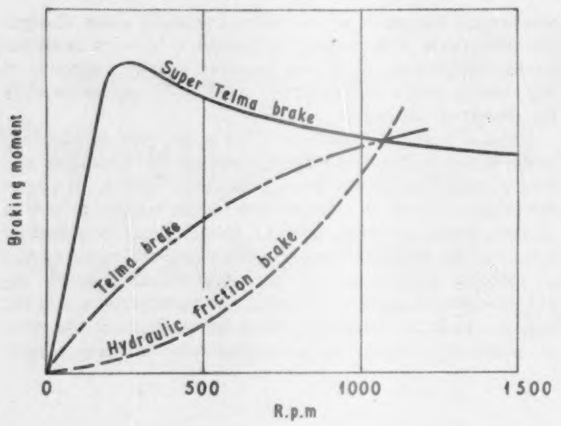


Fig. 7. Characteristic braking moment curves for normal friction type, Telma and Super Telma transmission brakes

(*After Endres. Reproduced by permission of V.D.I.-Verlag (3) G.m.b.H.*)

Ashanco, Saurer and Oetiker are examples of the exhaust brake; the sliding valve of the last mentioned is most suitable for petrol engines, as it has been found that, with heavily leaded fuels, lead deposited on the edges of the butterfly valves of other types soon prevented them from closing correctly.[6]

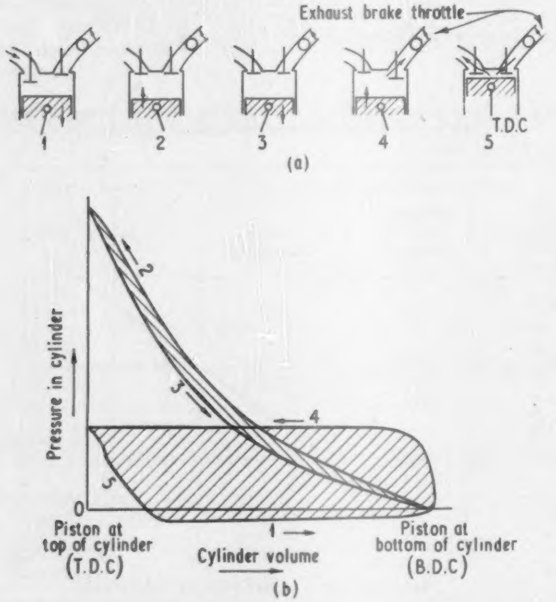
It is necessary when using an engine brake to ensure that the fuel supply to the cylinders is cut off completely or at least sufficiently to ensure that the normal firing stroke does not occur. At the same time free induction of air must be permitted if maximum efficiency is to be obtained. Webb and Lavender[7] have shown in laboratory tests that the tractive resistance of an engine with open induction and an exhaust brake can be over 80 per cent of the tractive power but that with the induction closed the tractive resistance is little better than that of the engine alone. Cutting off the fuel supply means that if the engine is disconnected from the driven wheels, as is necessary when changing gear, it is likely that the engine will stall and re-engagement of the drive may not be possible on a down gradient. Endres[3] considers that this makes it necessary to fit an interlocking device to ensure that the exhaust brake is released before gear changing, while Swiss operators who use engine brakes extensively instruct drivers when changing down on a descent to apply the handbrake sufficiently to take over from the engine brake, which is then released while the gear change is being made.

Fig. 8 shows the engine cycle and theoretical pressure/volume diagram for an engine fitted with an exhaust brake. It will be seen that during the valve overlap period when both inlet and exhaust valves are partly open, pressure equalization will occur by blow-back through the valves. This will also happen should the pressure in the exhaust pipe, which is common to all cylinders of the engine, exceed the exhaust valve spring pressure on any cylinder during its inlet stroke. The exhaust valve spring pressure is therefore in practice the limiting factor in the performance of the exhaust brake[6], although at least one device to overcome this limitation has been developed[8]. Owing to the blow-back, the design of the air intake cleaner should be considered when fitting an exhaust brake. Those of the oil-bath type are liable to spray oil over the engine.

The maximum deceleration achieved when using an exhaust brake without assistance from the main brakes is comparatively low, being about $0{\cdot}05\,g$ to $0{\cdot}09\,g$ between 30 and 20 m.p.h. (see Table II). However, considerable savings in brake lining wear, varying from 25 per cent to 50 per cent, have been reported [9][10] and overheating of the normal brake shoes on long descents has been prevented,

Fig. 8. Diagram of a 4-stroke engine cycle with exhaust brake in operation (a) and the equivalent pressure/volume diagram (b). The cycle is: inlet 1, compression 2, expansion 3, compression against exhaust brake 4, and pressure equalization 5

(*After Meyer. Reproduced by permission of the Society of Automotive Engineers (12).*)



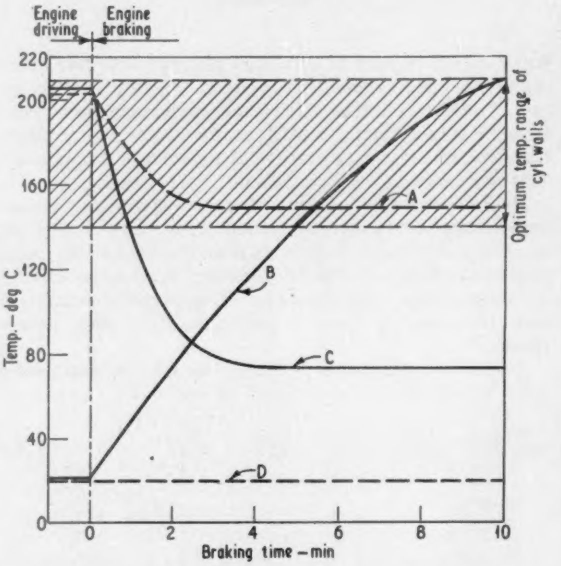


Fig. 9. Cylinder wall and brake shoe temperatures during descent, with and without an exhaust brake, of an air-cooled 4-stroke diesel (after Johannis)

A—Cylinder wall temperature during descent with exhaust brake
B—Brake shoe temperature during descent without exhaust brake
C—Cylinder wall temperature during descent without exhaust brake
D—Brake shoe temperature during descent with exhaust brake

(*Reproduced from "Automobiltechnische Zeitschrift," by permission of Franckh'sche Verlagshandlung (2).*)

see Fig. 9. The actual saving in the use and, therefore, wear of the wheel brakes will depend not only on the deceleration and degree and proportion of gradients on the operating route, but also on the type of auxiliary brake control used. This can be a separate hand or foot control, operation of which is at the discretion of the driver, or it can be automatic. In the latter case, the exhaust brake is put into operation by a switch that "makes" either when the driver removes his foot from the accelerator pedal or when he begins to depress the brake pedal, that is, before braking action is obtained with the main brakes. To obtain the maximum advantage from the auxiliary brake it is obvious that one of the automatic methods should be used. It has been found[11] that where application is at the discretion of the driver it tends to be ignored.

Johannis[2] has shown in tests with an air-cooled diesel engine that not only does an exhaust brake relieve the wheel brakes but it assists in maintaining the optimum, and, therefore, most economical engine temperature by preventing overcooling during long descents, see Fig. 9. This will be a greater advantage with an air-cooled than with a water-cooled engine, which does not cool down as quickly. He has also shown, Fig. 10, that the braking torque of an exhaust brake increases with engine speed, a further advantage over the friction brake.

It has been suggested[11][12] that the back pressure on the pistons during braking helps to prevent oil being drawn up past the piston rings and therefore wasted. Other advantages attributed to the exhaust brake are, a saving in fuel consumption, a reduction in driver fatigue and a possible saving in engine wear[11] due mainly to the fact that it permits a higher gear to be used for descents[3].

Advantages and limitations

Apart from the main advantages of reducing the work required from the main brakes on long descents, so leaving the main brakes in a good condition to meet emergencies, it is generally recognized that there is an economic gain since the life of the main brake friction linings is lengthened and they will require less frequent adjustment[9][10]. It is emphasized, however, that auxiliary brakes are not substitutes for wheel brakes. Working as they do on the driven wheels only, which in the majority of cases are the rear wheels, it is necessary to limit the amount of braking applied to prevent locking of the wheels in any circumstances, otherwise loss of vehicle control is very likely to occur[13]. The degree of braking which can safely be applied without doing this will depend on the type of vehicle, its weight and weight distribution, the condition and type of road surface, and its gradient. Endres[3] has deduced that, assuming a frictional coefficient of 0·2 between road and tyre, the limiting gradient for certain vehicles of the following types would be: for buses (single deck) 1 in 7 slope, for two- or three-axled vehicles (without trailer) 1 in 14 slope, and for an unladen lorry with a laden trailer (the worst combination when braking occurs only on the rear wheels of the prime mover) 1 in 25 slope. It should be remembered when considering the braking of vehicles descending slopes that the deceleration, compared with that obtained on a level surface, will decrease as the gradient increases. Fig. 11 shows a typical theoretical curve of the reduction in deceleration with increase in gradient[3]; this, however, is based only on the extra gravitational effect on the vehicle and does not take into account any load transfer from rear to front which could cause an even greater reduction.

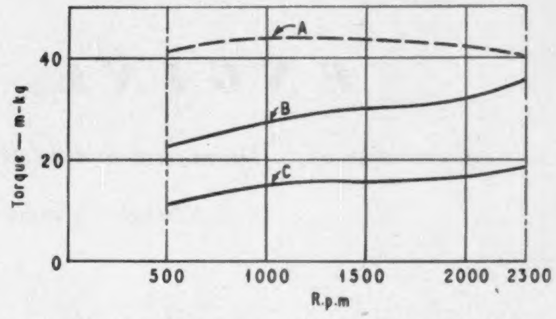
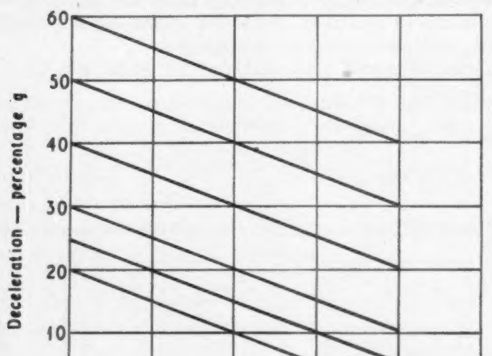


Fig. 10. Increase in braking torque of air-cooled 4-stroke diesel engine, with and without exhaust brake, and at various engine speeds (after Johannis)

A—*Engine torque at full load*
B—*Braking torque with exhaust brake*
C—*Braking torque without exhaust brake*

(Reproduced from "Automobiltechnische Zeitschrift," by permission of Franckh'sche Verlagshandlung(2).)

Fig. 11. Equivalent deceleration for various down gradients
(After Endres. Reproduced by permission of V.D.I.-Verlag, G.m.b.H.(3).)



References

1. R. D. LISTER, and R. G. STEVENSON. "Fifth Wheel for Measuring Speed and Braking Distance. *Bull. Mot. Ind. Res. Ass.*, 1952 (4), 18-22.
2. E. JOHANNIS. "The Effect of an Engine Brake on Thermal Condition of a Vehicle Diesel Engine." *Automobiltechnische Zeitschrift*, 1955, 57 (7), 202-4.
3. W. ENDRES. "Investigation on Additional Brakes for Heavy Lorries" ("Untersuchung uber die Talfahrtbremsen schwerer Strassenfahrzeuge"). *Deutsche Kraftfahrtforschung, Heft* 86. Düsseldorf, 1955 (Deutscher Ingenieur—Verlag G.m.b.H.).
4. ANON. Brussels Show. *Auto. Engr.*, 1956, **46** (2), 54.
5. ANON. "Auxiliary Braking Systems." *Auto. Engr.*, 1957, **47** (7), 283.
6. J. L. KOFFMAN. "Exhaust Brakes for Diesel Lorries." *Auto. Engr.*, 1953, **43** (566), 218-20.
7. C. R. WEBB and J. G. LAVENDER. "A Comparative Analysis of Various Methods of Engine Braking." *Proc. Instn. Mech. Engrs.*, 1955-56.
8. W. GLAMMAN. "Differential Drives for Superchargers." *Auto. Engr.*, 1955, **45** (9), 376.
9. ANON. "Braking of Heavy Road Vehicles." *Engineering*, 1954, **177** (4595), 252.
10. ANON. "Ashanco Exhaust Brake." *Auto. Engr.*, 1954, **44** (2).
11. ANON. "Exhaust Brake for Buses." *Mot. Transp.*, 1953, **87** (2500), 6.
12. W. E. MEYER. "Trucks Could Use Manifold Braking in Addition to Wheel Braking." *J. Soc. Automotive Engrs.*, 1955, **63** (12), 20-1.
13. H. J. H. STARKS. "The Factors which Influence the Braking Performance of Vehicles." *Bull. Mot. Ind. Res. Ass.*, 1950, (2), 17-33.
14. J. F. MOON. "Is Britain Behind in Braking?" *Comm. Mot.*, 1955, **102** (2626), 150-52.
15. ANON. "New Exhaust Brake." *Mod. Transp.*, 1954, **70** (1818), 13.

ENGINE BALANCE

Calculation of the Dimensions and the Moment of Inertia of the Balance Weights of a Single-Cylinder Reciprocating Unit

K. Weiss

CALCULATIONS to determine the dimensions and the moment of inertia of the balance weights necessary on a crank mechanism are, of course, based on details of the weights of the reciprocating and rotating masses. In single-cylinder engines, it is common practice to balance from 0·5 to 0·6 of the reciprocating and the full amount of the rotating masses. This, of course, is done by employing two balance weights, attached one to each crank.

W. H. Schneider, in an article entitled "Balancing of Reciprocating Masses," published in the July 1953 issue of *Australasian Engineer*, recommends that, to avoid over-balance, the value of the component balancing the reciprocating masses be found from the formula:

$$\frac{W_{rec}}{g}(0{\cdot}5+0{\cdot}36\lambda) \qquad (1)$$

In this formula, W_{rec} is the total weight of all the reciprocating masses, including the appropriate fraction of that of the connecting rod, and λ is the ratio of the crank radius to the connecting rod length.

For two balance weights, the following equation can, therefore, be used:

$$[W_{rec}(0{\cdot}5+0{\cdot}36\lambda)+W_{rot}]r=2W_{BW}x \qquad (2)$$

Where W_{rot} is the total weight of the rotating masses, which comprises that of the crank pin, the unbalanced parts of the crank webs and the appropriate fraction of the connecting rod, W_{BW} is the weight of one balance weight only, r is the crank radius and x is the distance between the centre of gravity of the balance weight and the axis of the crankshaft.

The centroid of a balance weight of simple form can be found by calculation, but for complicated forms, an experimental method is often used. Formulae have been derived by the author, which enable the rapid calculation of the necessary overall dimensions and the moment of inertia of a balance weight, the shape of which is represented by a portion of a sector of a circle with its centre point on the axis of the crankshaft, Fig. 1.

Calculation of the centre of gravity of a balance weight of uniform thickness and of the shape shown in Fig. 2.

Before finding the centre of gravity of the balance weight, the position of the centroid of the sector representing it must be calculated. If A_1 is the area of the triangle and A_2 the area of the complete sector, then (A_2-A_1) is the area of the balance weight. The difference of the moments of those areas, about point *0*, when divided by the area of the balance weight, will give the distance of the centroid from point *0*. Now, the equation for the moment is:

$$x_2A_2-x_1A_1=x(A_2-A_1) \qquad (3)$$

where x, x_1 and x_2 are the distances of the centroids of the three areas from point *0*.

But:

$$x_1=\frac{2}{3}h \qquad (4)$$

$$x_2=\frac{2}{3}r_0\frac{\sin\alpha}{\frac{\alpha}{57{\cdot}3}}$$

$$=38{\cdot}197\, r_0\frac{\sin\alpha}{\alpha} \qquad (5)$$

$$A_1=h^2\tan\alpha \qquad (6)$$

$$A_2=\frac{r_0^2}{2}\frac{2\alpha}{57{\cdot}3}$$

$$=\frac{r_0^2\alpha}{57{\cdot}3} \qquad (7)$$

Therefore:

$$x=\frac{\frac{2}{3}r_0^3\sin\alpha-\frac{2}{3}h^3\tan\alpha}{\frac{r_0^2\alpha}{57{\cdot}3}-h^2\tan\alpha} \qquad (8)$$

In the above equations, h is the distance between the axis of the crankshaft and the lower face of the crank web.

Calculation of the outer radius of the balance weight

For any given crank mechanism, the value of the left-hand side of equation (2) is a constant, which can be called C. Therefore:

$$C=2W_{BW}x \qquad (2a)$$

But $W_{BW}=(A_2-A_1)t\rho$, where t is the uniform thickness, in inches, of each balance weight and ρ the density of the balance weight material, in lb/in^3. Thus, equation (2a) can be written:

$$C=2(A_2-A_1)\,t\rho\,\frac{\frac{2}{3}r_0^3\sin\alpha-\frac{2}{3}h^3\tan\alpha}{(A_2-A_1)}$$

or

$$C=\frac{4}{3}t\rho\,(r_0^3\sin\alpha-h^3\tan\alpha)$$

Therefore:

$$r_0=\sqrt[3]{\left(\frac{3}{4}\frac{C}{t\rho}+h^3\tan\alpha\;\frac{1}{\sin\alpha}\right)}\ \text{in} \qquad (9)$$

Calculation of the moment of inertia of the balance weight, in Fig. 2, about an axis perpendicular to the plane through point 0

The polar moment of inertia required is the difference of the moments of inertia of the sector of the circle and the triangle, both about the axis through *0*.

(a) Moment of inertia of the sector of the circle, Fig. 3:

$$dI_{p2}=dAr^2$$

$$=2r\frac{\alpha}{57{\cdot}3}r^2dr$$

$$=\frac{2\alpha}{57{\cdot}3}r^3dr$$

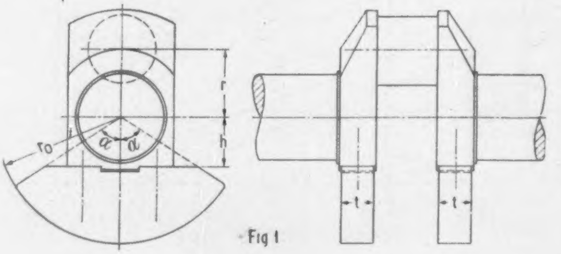


Fig 1

Therefore:

$$I_{p2}=\int_0^{r_0}\frac{2\alpha}{57\cdot3}r^3dr$$

$$=\frac{2\alpha}{57\cdot3}\frac{r_0^4}{4} \qquad (10)$$

(b) Moment of inertia of the triangular area, Fig. 4:
The polar moment of inertia of the triangular area about the axis through *0* is the sum of the moments of inertia about two perpendicular axes intersecting at the point *0*.

$$dI_{yy}=2x\tan\alpha\, dx\, x^2$$
$$=2\tan\alpha\, x^3\, dx$$

$$I_{yy}=\int_0^h 2\tan\alpha\, x^3\, dx$$

$$=\frac{h^4}{2}\tan\alpha$$

$$dI_{xx}=2\left(h-\frac{y}{\tan\alpha}\right)dy\, y^2$$

$$=2\, hy^2dy-\frac{2y^3}{\tan\alpha}\, dy$$

$$I_{xx}=\int_0^{h\tan\alpha}\left(2hy^2-\frac{2y^3}{\tan\alpha}\right)dy$$

$$=\frac{2}{3}h^4\tan^3\alpha-\frac{2h^4}{4}\tan^3\alpha$$

$$=\frac{1}{6}h^4\tan^3\alpha$$

$$I_{p1}=I_{xx}+I_{yy}$$

$$=\frac{h^4}{6}\tan^3\alpha+\frac{h^4}{2}\tan\alpha$$

$$I_{p1}=\frac{h^4}{2}\tan\alpha\left(1+\frac{\tan^2\alpha}{3}\right) \quad \text{in}^4 \qquad (11)$$

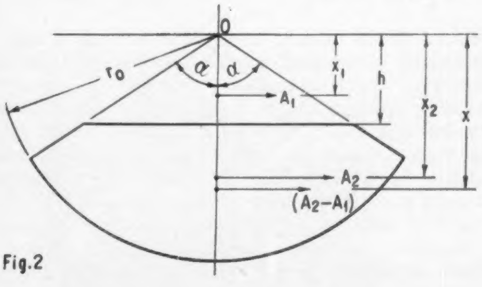


Fig.2

Therefore, the polar moment of inertia of the balance weight area is:

$$I_{p2}-I_{p1}=\frac{\alpha}{57\cdot3}\frac{r_0^4}{2}-\frac{h^4}{2}\left(1+\frac{\tan^2\alpha}{3}\right)\tan\alpha \quad \text{in}^4 \qquad (12)$$

and the moment of inertia of the balance weight, assuming that it is of uniform thickness and density, is:

$$I_{BW}=\left[\frac{\alpha}{57\cdot3}\frac{r_0^4}{2}-\frac{h^4}{2}\tan\alpha\left(1+\frac{\tan^2\alpha}{3}\right)\right]\frac{t\rho}{g} \quad \text{lb-in-sec}^2 \qquad (13)$$

where $g=386$ in/sec^2.

If α is taken as 60 deg, and the balance weights are of cast iron, equations (9) and (13) become:

$$r_0=\sqrt[3]{\left(\frac{3}{4}\frac{C}{0\cdot28t}+h^3\, 1\cdot732\right)\frac{1}{0\cdot866}}$$

Or:

$$r_0=\sqrt[3]{3\cdot09\frac{C}{t}+2h^3} \qquad (9a)$$

and $$I_{BW}=\left[0\cdot524\, r_0^4-1\cdot732\, h^4\right]\frac{t}{1{,}378} \qquad (13a)$$

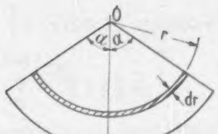


Fig.3

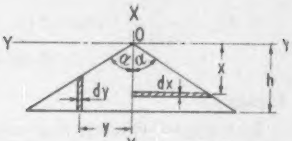


Fig.4

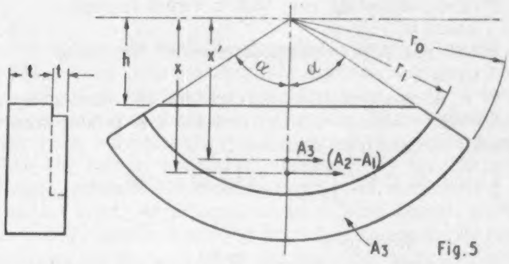


Fig.5

Calculation of the outer radius of a balance weight of non-uniform thickness, Fig. 5.

For design purposes, it is sometimes inconvenient for the balance weight to be of uniform thickness. In this case, the method already described can still be used as follows, to calculate the outer radius and the moment of inertia of the weight. Each balance weight can be assumed to consist of two parts, one as shown in Fig. 2 and the second a ring sector. Equation (2a) may therefore be written as:

$$C=2(W_{BW}\, x+W_{BW}'\, x') \qquad (14)$$

but $W_{BW}'=A_3\, t'\rho$

$$=\frac{\alpha}{57\cdot3}(r_0^2-r_i^2)\, t'\rho \qquad (15)$$

Therefore:

$$x'=\frac{\frac{2}{3}\sin\alpha\,(r_0^3-r_i^3)}{\frac{\alpha}{57\cdot3}(r_0^2-r_i^2)} \qquad (16)$$

and $$C=2\left[\left(\frac{2}{3}r_0^3\sin\alpha-\frac{2}{3}h^3\tan\alpha\right)t\rho+\frac{2}{3}\sin\alpha\,(r_0^3-r_i^3)t'\rho\right]$$

Therefore:

$$r_0=\sqrt[3]{\frac{1}{t+t'}\frac{3C}{4\rho\sin\alpha}+\frac{t\, h^3}{\cos\alpha}+t'\, r_i^3} \qquad (17)$$

In equation (17), t' is the thickness, r_i the inner radius and x' the distance of the centroid of the ring sector from the axis of the crankshaft. Values of t' and r_i can be chosen to satisfy the design requirements of the crank mechanism.

Calculation of the moment of inertia of the balance weight of a shape as shown in Fig. 4.

The total moment of inertia of each balance weight about an axis through point *0* is the sum of the moments of inertia of the two parts.

$$I_{BW}=\left[\frac{\alpha}{57\cdot3}\frac{r_0^4}{2}-\frac{h^4}{2}\tan\alpha\left(1+\frac{\tan^2\alpha}{3}\right)\right]\frac{t\rho}{g}$$
$$+\frac{\alpha}{57\cdot3}\frac{r_0^4-r_i^4}{2}\frac{t'\rho}{g}$$
$$=\frac{\rho}{2g}\left[\frac{\alpha}{57\cdot3}r_0^4(t+t')-h^4\tan\alpha\left(1+\frac{\tan^2\alpha}{3}\right)t\right.$$
$$\left.-\frac{\alpha}{57\cdot3}r_i^4\, t'\right] \quad \text{lb-in-sec}^2 \qquad (18)$$

If $\alpha=60$ deg, $\rho=0\cdot28$ lb/in^3 and $g=386$ in/sec^2, equation (17) can be rewritten:

$$r_0=\sqrt[3]{\frac{1}{t+t'}(3\cdot09C+2t\, h^3+t'\, r_i^3)} \quad \text{in} \qquad (17a)$$

and

$$I_{BW}=\frac{1}{2{,}756}\Big[1{\cdot}045\,r_0^4\,(t+t')-3{\cdot}464\,h^4t-1{\cdot}045\,r_i^4\,t\Big]$$
lb-in-sec² (18a)

Example

In a single-cylinder, four-stroke, high-speed diesel engine of 4 in bore and 5¼ in stroke, the weights are as follows:

Alloy piston assembly, including gudgeon pin	4 lb
Forged connecting rod, with a centre-to-centre length of 11·8 in	6·6 lb
Crank pin plus unbalanced parts of the crank webs	9·6 lb

If it is assumed that one-third of the connecting rod reciprocates with the piston and the rest rotates with the crank pin, then from equation (2):

$$\left[6{\cdot}6\left(0{\cdot}5+0{\cdot}36\,\frac{2{\cdot}625}{11{\cdot}8}\right)+14\right]2{\cdot}625=2W_{BW}x$$
$$=C$$
$$C=46{\cdot}80 \text{ in-lb}$$

For a cast iron balance weight, from equation (9a), where $\alpha=60$ deg, $t=1{\cdot}25$ in and $h=2$ in:

$$r_0=\sqrt[3]{3{\cdot}09\,\frac{46{\cdot}80}{1{\cdot}25}+2\times8}$$
$$=\sqrt[3]{132}$$
$$=5{\cdot}1 \text{ in}$$

From equation (13a), the moment of inertia is:

$$I_{BW}=\Big[0{\cdot}524\times673-1{\cdot}732\times16\Big]\frac{1{\cdot}25}{1{,}378}$$
$$=\Big[352-27{\cdot}8\Big]\frac{1{\cdot}25}{1{,}378}$$
$$=0{\cdot}294 \text{ lb-in-sec}^2.$$

Assuming the balance weight to be similar in shape to that of Fig. 5, where $t'=\frac{1}{2}$ in and $r_i=4$ in, r_0 can be found by using equation (17a):

$$r_0=\sqrt[3]{\frac{1}{1{\cdot}75}\left(3{\cdot}09\times46{\cdot}8+2\times1{\cdot}25\times8+\frac{1}{2}\times64\right)}$$
$$=\sqrt[3]{112{\cdot}5}$$
$$=4{\cdot}83 \text{ in}$$

and, from equation (18a):

$$I_{BW}=\frac{1}{2{,}756}\Big[1{\cdot}045\times542\times1{\cdot}750-3{\cdot}464\times16\times1{\cdot}25-1{\cdot}045\times256\times\frac{1}{2}\Big]$$
$$=\frac{1}{2{,}756}\Big[990-79{\cdot}28-134\Big]$$
$$=0{\cdot}282 \text{ lb-in-sec}^2.$$

Conclusion

A solution for the outer radius of the balance weight can be readily obtained by application, to the formulae, of the crankshaft dimensions and the masses to be balanced. The thickness of the balance weight is usually equal to or slightly smaller than that of the crank web. If, for design purposes, the outer radius is predetermined, equations (9) or (17) can be used to calculate the thickness of the balance weights. Then, the moment of inertia can be found from equations (13) or (18). The method described has the advantage that, in the final calculation, the finding of the centroid can be by-passed and this greatly reduces the time required to make the calculations.

Position Identification for Data Control

IN view of the increasing interest in data control systems for machine tool operation in this country, in the United States of America and in Germany, we wish to draw attention to a series of articles on American developments appearing in our associated journal, *Aircraft Production*, beginning with the October issue. The first article, on which these notes are based, deals with position identification. In any data control system for the operation of machine tools, the position identifying device is of fundamental importance, since it provides the feedback information specifying the displacement of the individual slides from a datum position. Precision in machining depends, initially, on the magnitude of the smallest movement that can be controlled with repeatable accuracy; hence, the importance of the position identifying device.

Owing to the character of the data handling equipment, it is desirable that the feedback information should be of electrical nature, or at least be suitable for easy conversion into electrical form. Because of this, most systems use either entirely electrical devices or combined photo-electric units.

The position identifying unit may be suitable for monitoring the displacement of a machine slide at one pre-established setting, or it may be designed to provide continuous information of the displacement of the slide. In the latter case, it is suitable for continuously controlled contour machining operations. If a position identifying device can provide signals suitable for use with a system of continuous control, then, provided the degree of dimensional resolution is of the correct order, it can also be used as the feedback unit in a co-ordinate setting system. The converse is not always necessarily true. Hitherto, the discriminatory power of position identification feedback units has been higher in those developed for co-ordinate setting systems than in those for continuously controlled contour machining. The most frequent use of data controlled setting systems is in applications to jig boring machines on which very close limits must be maintained. More recently, continuous control systems have been designed for installation on standard milling machines for use on work with much wider permissible tolerances than can be allowed in jig boring.

The first article in the *Aircraft Production* series describes in detail an American position identifying device, the Farrand Inductosyn, which has a high power of resolution and can be used as the information feedback unit in either form of data control system.

The equipment that must be used in these systems will be unfamiliar to both production and maintenance staffs in machine shops. Because of this, these very detailed and authoritative articles, which result from an American tour undertaken specially for the purpose, will be of great interest and use to engineers who will eventually have to deal with data control systems.

Welding Nuts

Automatic Feeding Equipment for Better and Faster Production

IN the conventional set-up for attaching welding nuts to steel panels, the operator has to carry out three functions: first, he must manipulate the sheet steel panel into the resistance welding machine and locate it in relation to the electrodes; second, he must pick up a nut and place it accurately on the panel under the electrode, while supporting and balancing the panel; third, he must operate the welding machine after withdrawing his hand from the welding nut and while retaining control of the panel. To achieve a more efficient operating cycle and to improve safety in the production of motor car body panels, Briggs Motor Bodies Ltd. commissioned Rhoden Partners Ltd., Design and Development Engineers, 51 North Row, London W.1, to develop automatic feeding equipment. The resulting equipment, mounted on a standard 50 kVA "British Federal" air-operated resistance welding machine, is shown in Fig. 1.

The equipment includes a standard "Syntron" bowl feeder that orientates the nuts in the correct welding position and delivers them in a row to an attachment in the form of a breech block. When the top electrode reaches its upper position in the operation sequence, an air-cylinder in the breech block transfers one nut to the entrance of the feed tube, whence compressed air delivers the single nut through the tube into the jaws of the electrode assembly. After the operator has placed the panel in position on a locating pin in the bottom electrode, he depresses a pedal to initiate the welding cycle. The top electrode then descends and carries the nut in the jaws down until it registers on the locating pin and contacts the panel. The jaws, shown in Fig. 2, then open automatically, well clear of the welding area, while the nut is clamped to the panel by the electrode. Forging pressure and welding current are then applied for a pre-set time. At the end of the welding period, the top electrode is retracted and the jaws close automatically ready to receive the next nut when the electrode reaches its upper position.

The mechanisms for transporting a single nut and for opening and closing the jaws are timed off the basic welding cycle. They derive their motions from the original mechanical and pneumatic operations, so that no additional controlling or working movements have to be carried out by the operator. This new feeding equipment in no way reduces the capacity of the machine, nor does it obstruct the operator's view. Since the operator has both hands free to control the panel, he can do so more quickly and more reliably. Furthermore, both hands are kept away from the electrodes at all times. In this way, the cycle time is reduced, especially since the time required for the return stroke of the top electrode, hitherto a purely idle movement, is now utilized for receiving a nut in the jaws of the nut holder.

Fig. 1. Resistance welding machine with automatic feeding equipment for nuts

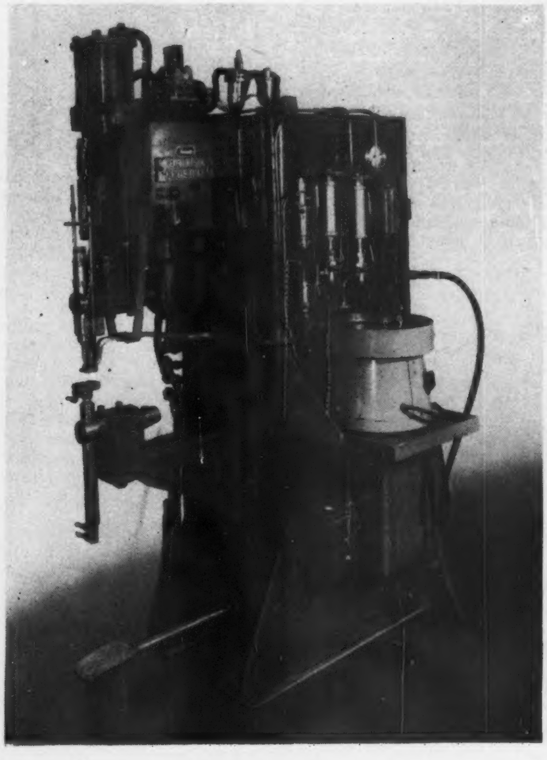
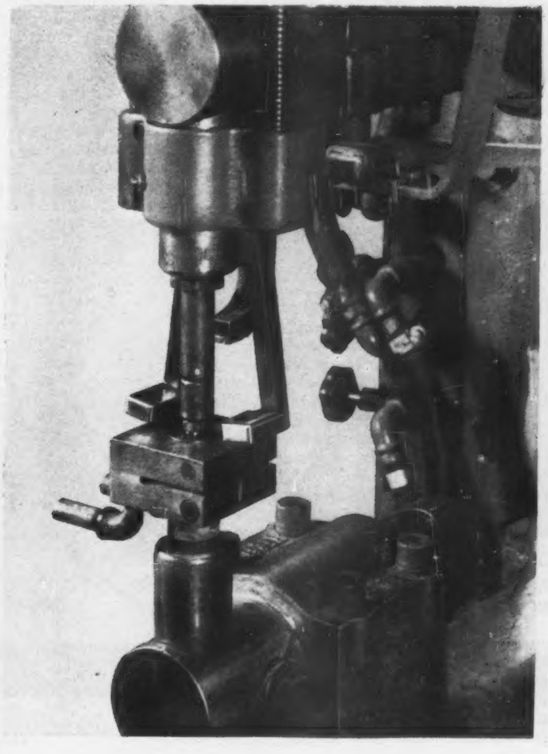


Fig. 2. Close-up showing the delivery end of the pneumatic system and the nut-retaining jaws



CURRENT PATENTS

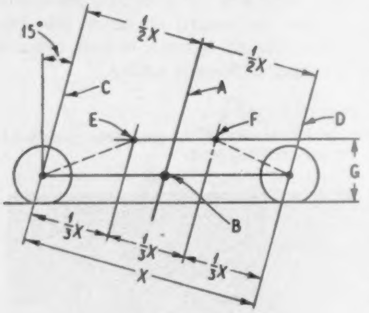
A REVIEW OF RECENT AUTOMOBILE SPECIFICATIONS

Seating arrangement

This invention aims at the formulation of a specific location for the seats in a motor vehicle, to secure maximum comfort at the minimum structural expenditure, on the essential basis of the dynamics of the vehicle.

In the diagram, A indicates a transverse medial plane, inclined at 15 deg to the vertical and intersecting at B a horizontal plane containing the front and rear axles. The angle of 15 deg represents a mean value of the directions of the resultants of shocks experienced by the vehicle when passing over road irregularities. Planes C and D, passing through the wheel centres, are parallel to plane A and the distance between them constitutes the basis X for the seat arrangement. A typical mean value for this basis is 2·55 metres.

The centre of mass E of the loaded front seat and the centre of mass F of the loaded rear seat are equally spaced from plane A and the distance between them is equal to the distances from adjacent planes C and D. Considerations of the shape of the seats and the required road



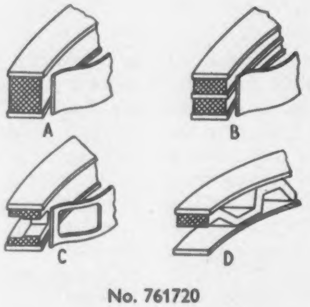
No. 763772

clearance will determine the height G of the horizontal plane containing the centres E and F. *Patent No.* 763772. *Daimler-Benz A.G.* (*Germany*).

Composite piston rings

Claimed to be an improvement on either the solid cast iron ring or the steel laminate ring, this ring comprises at least two cast iron or steel laminations spaced by one or more inserts of a synthetic resin. These inserts, which have no sealing function, may be non-elastic or slightly elastic and may be separate from the laminations or securely bonded to them. A synthetic resin capable of withstanding the temperatures experienced in internal combustion engine cylinders is polytetrafluoroethylene, a commercially available version of which is "Teflon". In order to increase the rate of heat conduction, the p.t.f.e. may be compounded with a metal powder. . .

Constructional examples show at A a ring consisting of two steel laminations spaced by an insert of synthetic resin. The steel laminations are constrained to the cylinder wall by a wave-shaped expander ring. At B is a ring having three steel laminations. The low specific weight of



No. 761720

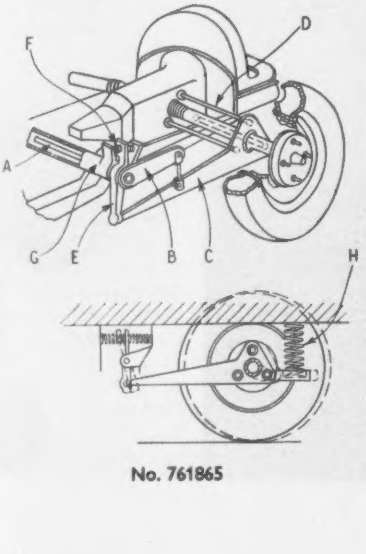
the synthetic resin makes it possible to produce a ring outside the pulsation range.

Oil scraper rings may have an apertured p.t.f.e. insert, as at C, and be expanded by a perforated spring. Alternatively, the insert may have a wave-like form, as at D. This shaping has the property of resilience in the axial direction and consequently discrepancies in ring groove width are automatically compensated. *Patent No.* 761720. *H. Teves and E. A. Teves* (*Germany*).

Independent rear suspension

In a rear suspension embodying swinging half-axles guided by longitudinally disposed swinging strut members, horizontal shocks are liable to be transmitted to the vehicle chassis at the strut pivots. Rubber bushings at these points are not, it is suggested, completely effective in eliminating shock transmission as they must be heavily pre-tensioned. It is arranged, therefore, for the strut member to be permitted a limited horizontal movement against the constraint of restraining springs.

For a construction in which the main suspension spring is a transversely arranged torsion bar A, the free end of the bar carries a lever arm B pivotally linked to the strut C secured to the swinging half-axle which is ball-jointed to the transmission casing. The wheel is guided by three



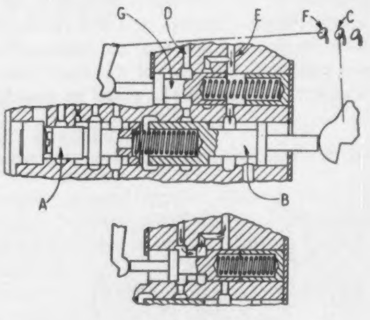
No. 761865

members D, connected to both the casing and the strut C by ball and socket joints. At its end remote from the axle, the strut C is connected to a double-armed lever E by a ball joint. Lever E is pivotally mounted concentric with the torsion bar and is biased to a central, vertical position by helical springs F bearing against abutment lugs on the transverse member G enclosing the torsion bar.

A second example shows a similarly mounted half-axle with a helical main suspension spring H and a double-armed lever pivoted on a frame bracket. Movement of the wheel and the lever under horizontal shock load is indicated in broken outline. *Patent No.* 761865. *F. Porsche* (*Germany*).

Control system for automatic transmission

In vehicles equipped with an automatic transmission, control is normally by means of an accelerator pedal and a brake pedal.



No. 762370

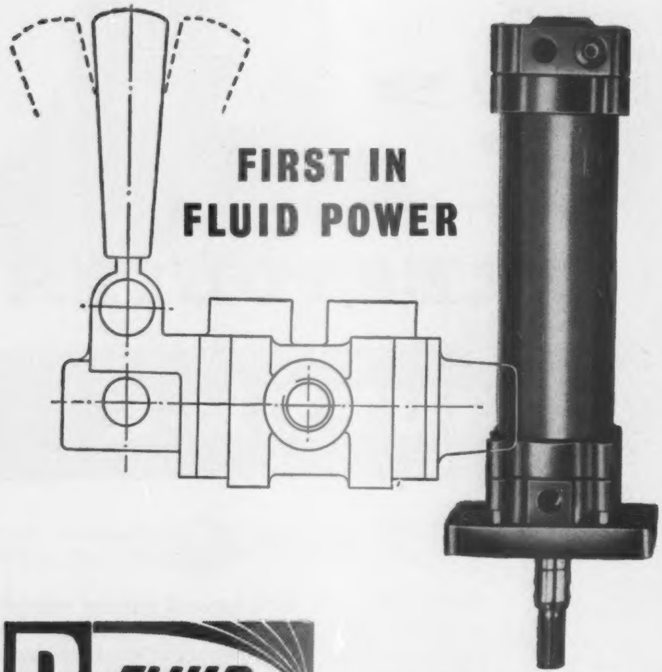
It is usually arranged that when particularly good acceleration is required a change down to a lower gear is effected by depressing the accelerator pedal beyond the normal fully opened position. With this so-called "kick-down" method of control such a change can be made only at full throttle. As an improvement on this method the invention provides an additional control pedal which independently operates a valve mechanism that modifies the change-controlling fluid pressure to effect engagement of the next lower gear.

The control illustrated is suitable for the transmission described in Patent No. 522881. This includes a fluid-pressure operated control incorporating a piston valve A which is spring-loaded in conjunction with a second piston valve B which is movable in accordance with the position of the accelerator pedal C. Piston valve B controls the supply of pressure fluid from duct D to duct E leading to a mechanism effecting a change to a lower gear on the "kick-down" of pedal C.

The additional change-down pedal F actuates a third piston valve G controlling a by-pass duct interconnecting ducts D and E, irrespective of the position of the accelerator pedal C. *Patent No.* 762370. *Rolls-Royce Ltd.*


HOFFMANN
Performance amply justifies our claim of complete reliability
BALL AND ROLLER BEARINGS
THE HOFFMANN MANUFACTURING CO. LTD., CHELMSFORD, ESSEX

Versatile
—Powerful
—for heavy duty on a wide range of work
Archdale
34" VERTICAL
REVERSIBLE AUTOMATIC FEEDS AND QUICK POWER TRAVERSE IN THREE DIRECTIONS
The characteristic power and rigidity of the 34" Vertical Milling Machine is demonstrated to good effect when taking a ¼" cut at 3½" per minute on this 4% carbon steel billet. This machine is installed at Geo. Swift & Sons Ltd., who find its great versatility invaluable in producing a wide range of machine tool components. All 12 feeds and spindle speeds are selected from the front of the machine, the speeds ranging from 29—520 or 36—638 R.P.M., and the feeds ¾" to 20" per minute.
JAMES ARCHDALE & CO., LTD.
BIRMINGHAM 16 · SOLE SELLING AGENTS : ALFRED HERBERT LTD. COVENTRY



FIRST IN FLUID POWER

Wherever steady, controlled movement is required oil-hydraulics really come into their own.
Most hydraulic equipment, however, is designed and built primarily for high-pressure operation. When used for low-pressure applications it runs at a fraction of capacity and it would therefore be far cheaper to design and build hydraulic equipment specially for low-pressure operation.
The new range of Baldwin low-pressure hydraulics meets this need. Right from the drawing-board stage this equipment keeps one aim in view—RELIABILITY. Polished hard-chrome plated cylinder bores and piston rods; self-adjusting oil, and water resistant seals; rugged malleable cast iron and steel construction, completely rust and corrosion proof inside and out: built-in cushioning to absorb the shock of heavy loads moving at high speeds, and many other quality design features ensure absolute efficiency even under the most arduous conditions. Unit construction enables Baldwin to offer 432 standard types of cylinder with 6 bore sizes and 8 standard mountings.
Ask any discriminating buyer, he will tell you that it *pays* to specify Baldwin low-pressure hydraulics.

Write now for your copy of this new Fluid Power Brochure G5-503, it gives complete easy-to-read information on the entire range of Baldwin cylinders, valves and accessories, for pneumatic and hydraulic use.

B FLUID POWER

BALDWIN FLUID POWER

One of the Harper Group of Companies

BALDWIN INSTRUMENT COMPANY LIMITED
DARTFORD · KENT · TEL: DARTFORD 2948

B37

HOW ARE THEY TINNING THIS JOB?

These 8′ 6″ diameter copper cylinders, produced by F. W. Hall Limited, of Norwich, are being tinned with FRYOLUX pure tin paint for use in a chemical works.

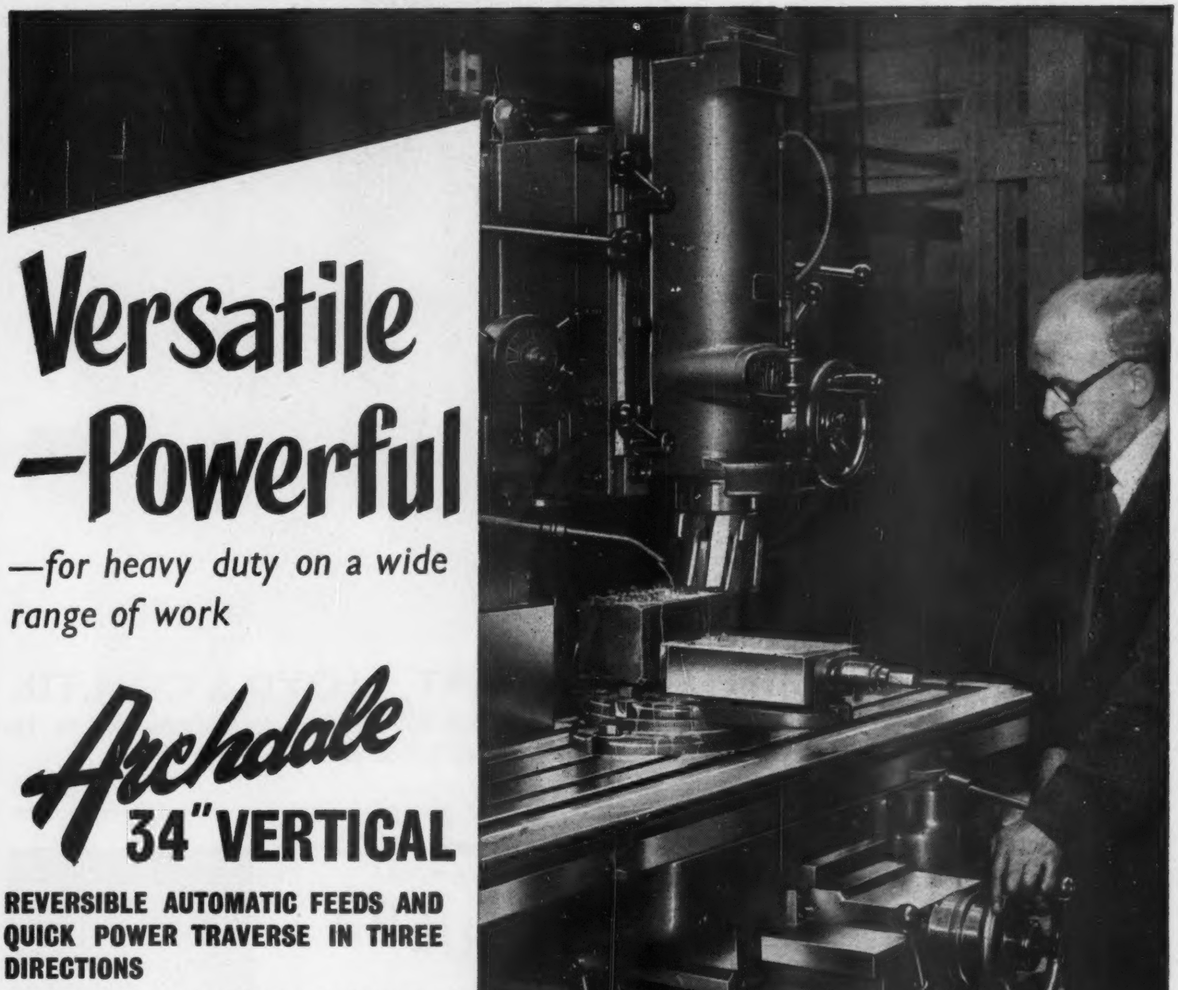
FRYOLUX is specially developed for all tinning and sweat soldering operations. It brushes easily and on application of heat it tins instantly, giving firm bond with base metal and covering large areas quickly and economically. *Please send for samples and prices. Export enquiries invited.*

.... with FRYOLUX pure tin paint

FRY'S Metal Foundries Limited

Tandem Works, Merton Abbey, LONDON, S.W.19. **Tel: MITcham 4023**

and at MANCHESTER · GLASGOW · BRISTOL · DUBLIN M.R.P. 67


Versatile
—Powerful
—for heavy duty on a wide range of work
Archdale
34" VERTICAL
REVERSIBLE AUTOMATIC FEEDS AND QUICK POWER TRAVERSE IN THREE DIRECTIONS
The characteristic power and rigidity of the 34" Vertical Milling Machine is demonstrated to good effect when taking a $\frac{1}{4}$" cut at $3\frac{1}{2}$" per minute on this 4% carbon steel billet. This machine is installed at Geo. Swift & Sons Ltd., who find its great versatility invaluable in producing a wide range of machine tool components. All 12 feeds and spindle speeds are selected from the front of the machine, the speeds ranging from 29—520 or 36—638 R.P.M., and the feeds $\frac{3}{4}$" to 20" per minute.
JAMES ARCHDALE & CO., LTD.
BIRMINGHAM 16 · SOLE SELLING AGENTS: ALFRED HERBERT LTD. COVENTRY

"This is a job for Coopers"

And it is in the design stage when the COOPER technicians can be of the greatest assistance. It is their business to know all there is to know about Filters and Strainers. They are prepared to design and manufacture to your own specific requirements.

COOPERS engineers are available for consultation at all times.

COOPERS

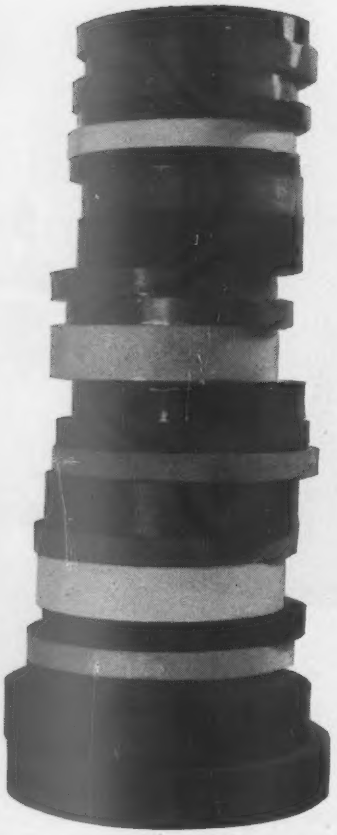
STRAINERS · FILTERS · GASKETS · WASHERS
LAMINATED SHIMS · PRESSWORK

COOPERS MECHANICAL JOINTS LTD., LLANFOIST WORKS, ABERGAVENNY **Telephone: 550**

A tape for EVERY use

Cellulose
P.V.C. (Unplasticised)
Cloth Sealing
Cloth Masking
Paper Masking
Waterproof Cloth
Super Strapping

PERMACEL Self-Adhesive Tapes



Sealing
Protecting
Masking
Waterproofing
Bundling
General Packing
Labelling
Watertight Packing
Heavy Duty Packing
Identifying
Export Packing

ILLUSTRATED LITERATURE FREE ON REQUEST FROM INDUSTRIAL DIVISION

PERMACEL

TRADE MARK

INDUSTRIAL MASKING TAPES

Manufactured by Johnson & Johnson (GT. BRITAIN) LTD. SLOUGH, BUCKS

WHERE ALL THE BEST

HOT BRASS PRESSINGS *COME FROM*

hPP

One of the SANBRA Group of Companies.

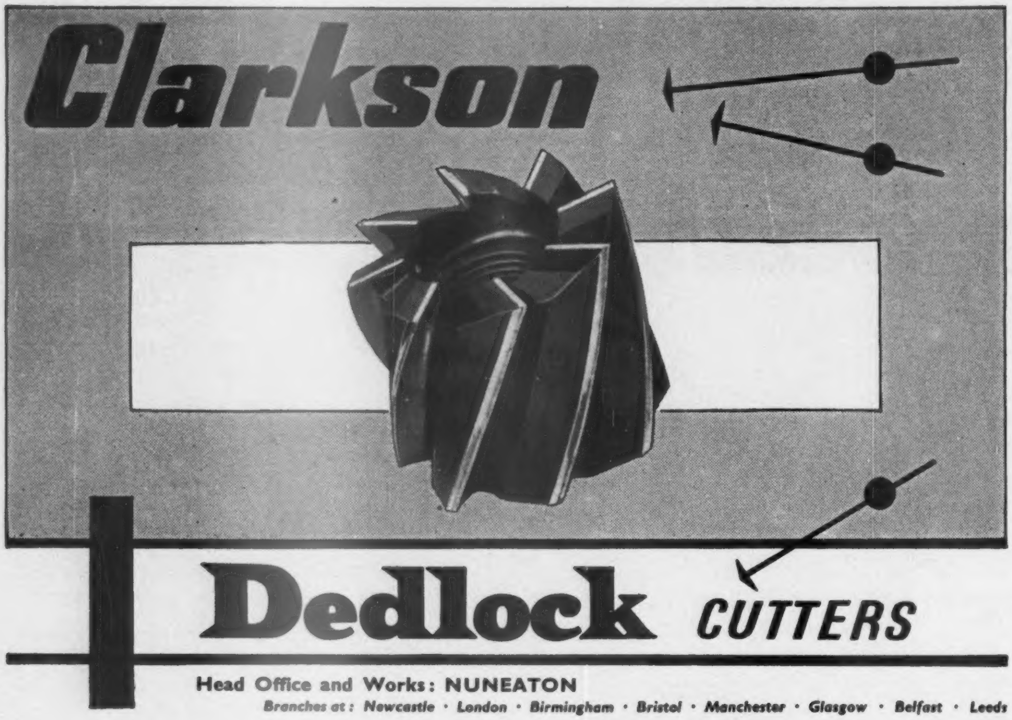
Here is a Works as modern as the minute, with the very latest in plant and equipment, planned to offer you the full benefits of specialised production—in the form of better quality, prompter deliveries, and keener prices. Whatever your needs in Hot Brass Pressings, Machined Pressings and Assemblies, you will find that H.P.P. can serve you better.

HOT PRESSED PRODUCTS

WHITEHALL ROAD, GREAT BRIDGE, TIPTON, STAFFS. *Telephone: TIPton 2731/2.*


CHURCHILL Quality... Service
available at every
CHURCHILL centre
CHARLES CHURCHILL & CO LIMITED
COVENTRY ROAD SOUTH YARDLEY BIRMINGHAM 25
BRANCHES LONDON GLASGOW NEWCASTLE MANCHESTER


FLEXIBILITY...
★ that only DOUBLE PROCESSED CREPED PAPER TAPE can give
Sellotape CREPED PAPER Masking Tapes meet all requirements of product finishing or special processing
INCLUDED IN THE SELLOTAPE RANGE:
No. 51 STANDARD
For most spraying work. Thin but extremely strong, ideal for masking sharp curves. Temperature-resistant up to 200°F for up to 2 hours—to 250°F for shorter periods. Non-discolouring on most painted and enamelled surfaces.
No. 575 HIGH TEMPERATURE
For stoving and drying processes. Adhesive remains firm up to 266°F for 12 hours.
No. 52 WAFER THIN
As No. 51 Standard, but extra thin and flexible. Prevents paint build-up.
No. 46
Specially designed for protection of rubber surfaces. Can be removed without leaving deposit after stoving up to 212°F.
* Double Processing—Absolute protection against spray penetration. Perfect keying of adhesive.
COATING
CREPE PAPER
COATING
ADHESIVE
Sellotape
REGD. TRADE MARK
MASKING TAPES
perfect start to a perfect finish
Sellotape is the Registered Trade Mark of Adhesive Tapes Ltd.
Sole Distributors in the U.K.—The INDUSTRIAL SELLOTAPE DIVISION of Gordon & Gotch Ltd., 8-10 Paul Street, London, E.C.2. Telephone: BIShopsgate 6511
G & G


Clarkson
Dedlock CUTTERS
Head Office and Works: NUNEATON
Branches at: Newcastle · London · Birmingham · Bristol · Manchester · Glasgow · Belfast · Leeds


you design . . .
we manufacture . . .
MECHANICAL RUBBER COMPONENTS for automobile & aircraft production
Send your "Problem piece" for our quotation
THE SHERBORNE RUBBER CO LTD
SHERBORNE STREET · BIRMINGHAM · 16 phone: EDG 3881·2·3

Unique

"Newallastic" bolts and studs have qualities which are absolutely unique. They have been tested by every known device, and have been proved to be stronger and more resistant to fatigue than bolts or studs made by the usual method.

A. P. Newall & Co., Ltd.

POSSILPARK GLASGOW · N

they go to Town ... *
at Marshall Sons and Co Ltd

In the toolroom, an A.E.I. type Radial is used for accurate work, in this case drilling a jig plate, the holes having been plotted previously on a jig boring machine.

* **FRED^K. TOWN & SONS, LTD** HALIFAX · YORKS

ESTABLISHED 1903

T11

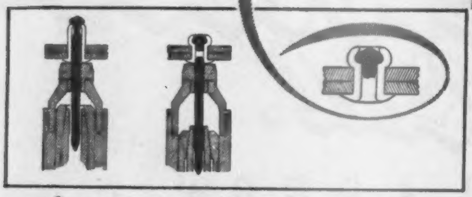
For quicker and more economical assembly

In constant use where speed, economy and simplicity are essential, the "Pop" Riveting System is solving assembly problems in an ever-increasing range of industries.

Almost limitless in its applications, it requires only one operation by one operator, working from one side of the material only, to set or clench the rivet, thereby enabling rivets to be set in positions inaccessible by other methods.

Our technical staff will be pleased to co-operate in solving your assembly problems.

The illustration above shows "Pop" Rivets being used attaching springs for holding moulding finishers—just one of the many applications in the motor industry.



TUCKER POP RIVETS

Geo. **TUCKER EYELET** Co. Ltd.

Walsall Road . . . **BIRMINGHAM 22**

Telephone: BIRchfields 5024 (7 lines) **Telegrams: EYELETS, BIRMINGHAM**

Consultants: **AIRCRAFT MATERIALS LTD. Midland Road, London, N.W.1**

"POP" is a regd. Trade Mark of Geo. Tucker Eyelet Co. Ltd.

Controlled Lubrication is good for you . . .

. . . and certainly *very good* for prolonging the life of your machinery.

The advantages of centralised systems for bearing lubrication are fully appreciated as an aid to production. Enots comprehensive range of lubrication equipment is adaptable to every industry and is indeed saving time, money and effort in the modern factory by reducing the number of breakdowns, eliminating costly repairs and adding years of service to the machine.

Send NOW!

Send now for these fully informative booklets which are free on request. Ask for them by number :—

OS 955. "One Shot" Lubricating System.
LS 952. "Benston" Lubricating System.
LGN 254. Lubricating Guns and Nipples, and
CF 154. Cutting Oil and Coolant Fittings.

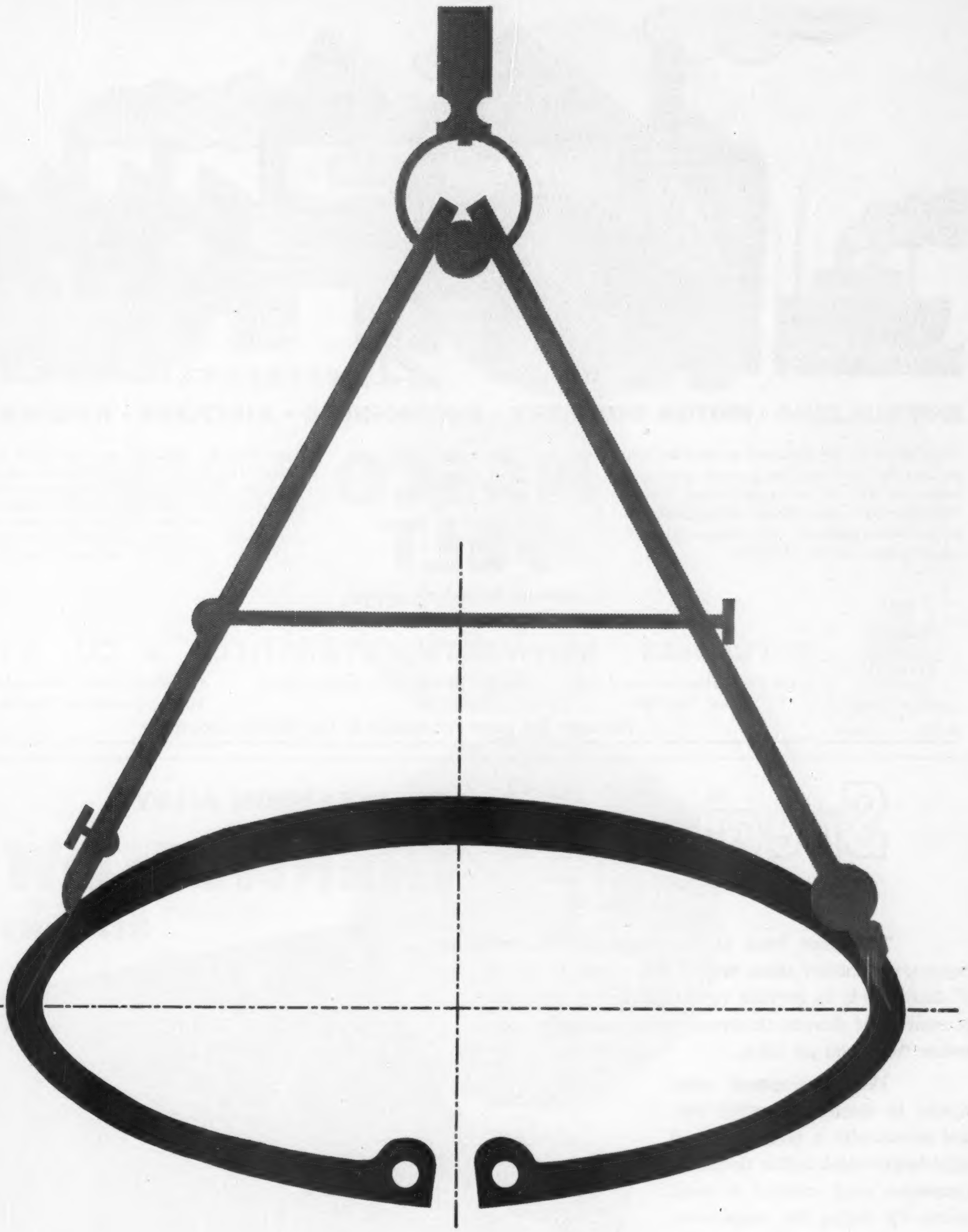
Leading manufacturers in the textile, printing, machine tool and engineering trades standardize on ENOTS Lubricating Equipment.

Enots

LUBRICATING EQUIPMENT

BENTON & STONE LTD. · ASTON BROOK STREET · BIRMINGHAM 6 · ENGLAND

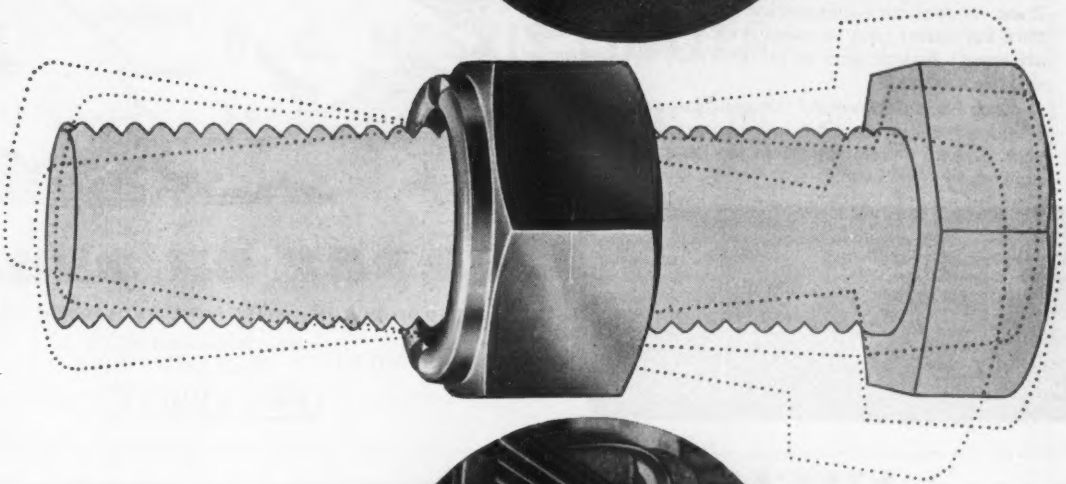
We shall be pleased to welcome you on Stand No. 24, Avenue O, 1st. Floor, MOTOR SHOW, Earls Court, October 16th—26th.



360° safe: SEEGER CIRCLIPS *of all sizes are designed*
and thrice tested ***for uniform pressure throughout their circumference***

Automotive Engineering Limited

One of the Sheepbridge Engineering Group
The Green, Twickenham, Middlesex
Telephone: POPesgrove 2206/9 Telegrams: MOTIF, Twickenham



Nothing rattles the Nyloc

Stubborn nuts, these Nylocs! Jerk hem, bump them, drench them, pelt them—the nylon insert holds them *firm*. Austins use Nylocs on the spring assembly of 5-and 7-ton trucks—on the U-bolts and shackle pins. These trucks are built for a tough life, but nothing shakes the Nyloc nuts loose. Nyloc self-locking nuts are also saving costly assembly time on the production line. And by using them in place of slotted nuts Austins have been able to fit grease nipples to *both* ends of each shackle pin. The all-metal Pinnacle nut and the Fibre nut with fibre insert can be just as helpful—and equally stubborn. Drop us a line and we'll gladly advise you which nut is best for *your* job.

FC

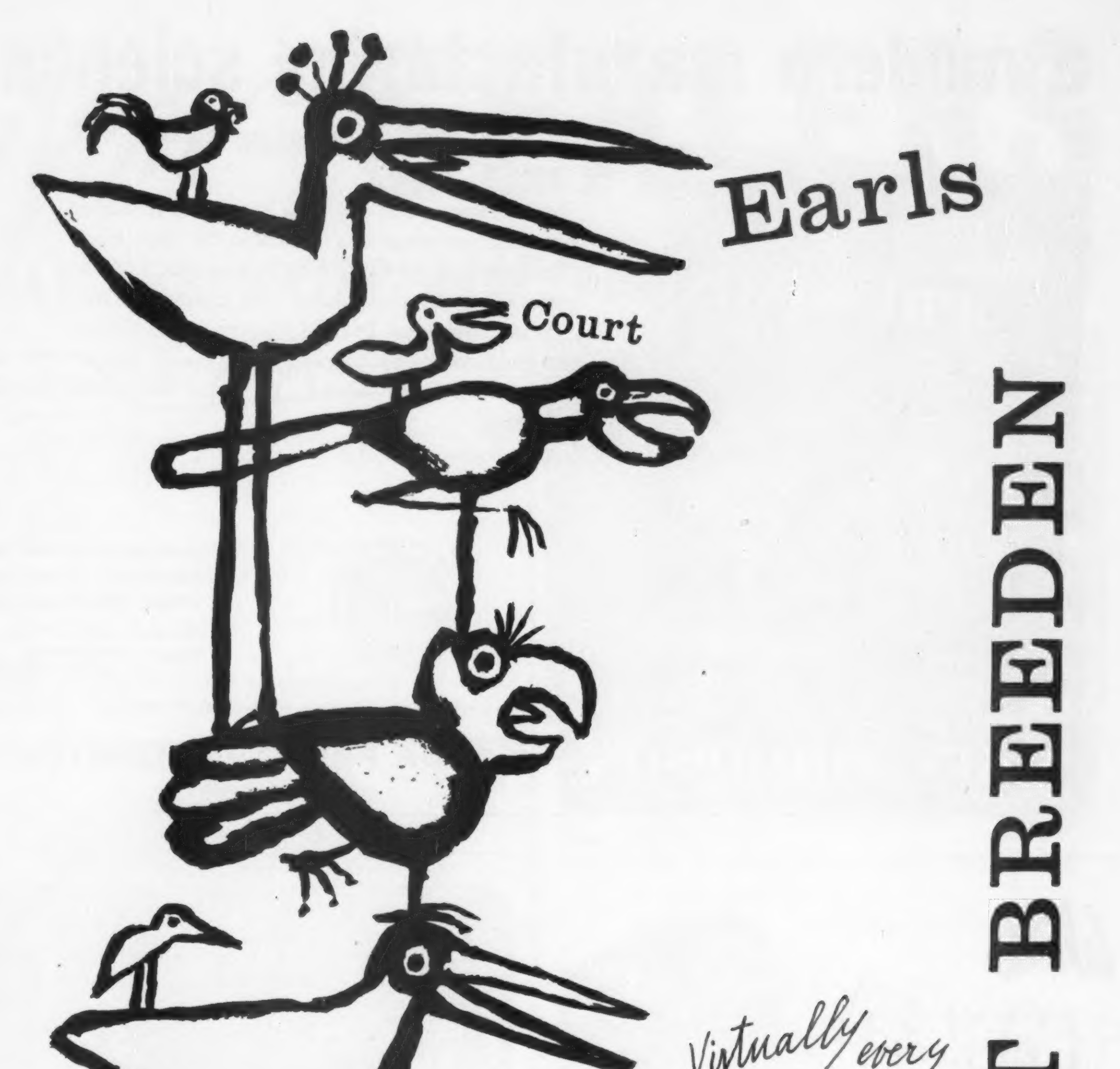
SIMMONDS AEROCESSORIES LIMITED

TREFOREST · PONTYPRIDD · GLAMORGAN

A MEMBER OF THE FIRTH CLEVELAND GROUP

Branches: London, Birmingham, Manchester, Glasgow, Stockholm, Copenhagen, Ballarat, Sydney, Johannesburg, Amsterdam, Milan and New York.

CRC24N


Earls
Court
Stand
328
Virtually every British car on the roads today carries components by
WILMOT BREEDEN
AF

Now—SEALERS

THAT ARE *WATERPROOF* AND REMAIN *PLASTIC*

GLASTICON '234' ALL-PURPOSE SEALER

Supplied in disposable cartridges for use in the GLASTIGUN. Each cartridge has its own nozzle which can be trimmed to regulate flow.

- *Adheres to all clean surfaces.*
- *Forms tough, tack-free surface skin.*
- *Can be overpainted with synthetic or cellulose paint.*
- *Available in black or natural colour.*

33/9 Per doz. Approx. 17 oz. net weight cartridges. One FREE GLASTIGUN with every 1 gross cartridges **or 17/6 each (both net trade).**

GLASTICORD '304' STRIP SEALER

Supplied on 12" reels in non-returnable containers.

- *Good adhesion.*
- *Heat-proof, moisture-proof, dust-proof.*
- *Remains plastic.*
- *Any section size strip available.*
- *Colours—black or buff. White on quotation.*

2/10 per lb. net trade.
(Weights and measures chart available.)

GLASTICON 'DUM-DUM' TYPE PUTTY

Supplied in tins and kegs.

- *Cannot harden or crack ; withstands vibration.*
- *Easily and cleanly applied.*
- *Used by the motor body industry for over 20 years.*
- *Available in black or, subject to special quotation, grey, brown or white.*

Net trade prices: **7 lb. tins 10/8 each.**
28 lb. kegs. 34/- each.
1 cwt. kegs 104/- each.

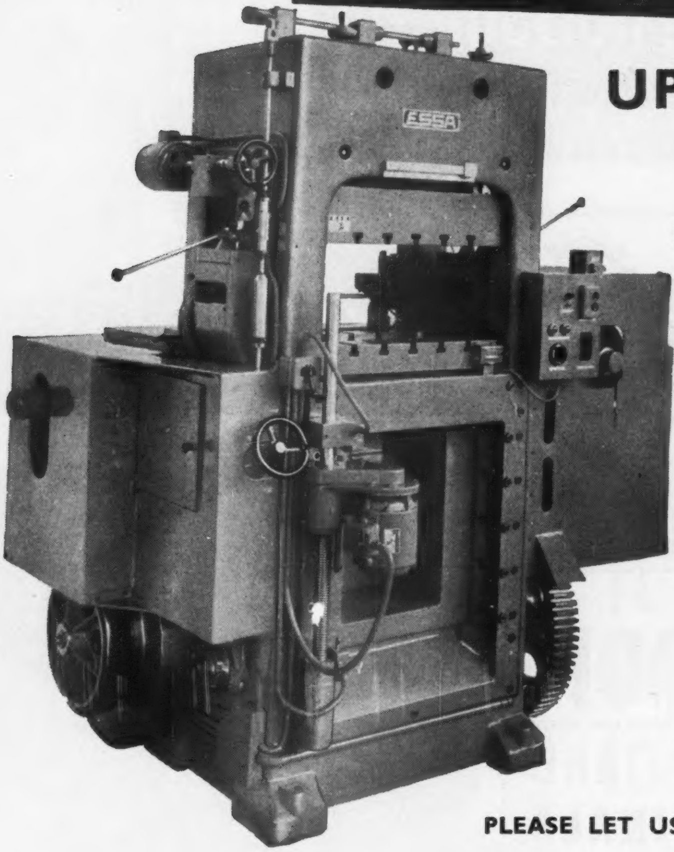
Available from all Motor Body Factors. Technical data and quotations direct from :

KELSEAL LTD.

MELLIER HOUSE, ALBEMARLE STREET, LONDON, W.1 Telephone: HYDE PARK 1411

Do you know why we work from the bottom upwards?

ESSA



UP-STROKING PRESSES

GIVE 7 TO 10 TIMES NORMAL TOOL LIFE—

PLUS GREATLY INCREASED PRODUCTION RATES ! ! !

This precise and rugged Swiss-built machine gives greatly extended tool life up to, invariably seven to ten times that obtained from any other type of press by virtue of its design whereby vertical backlash is entirely eliminated; this means that the punches can be set so that they never enter the die. The machine shown is a double crank type equipped with double roll feed, strip end shear, scrap shear, and automatic stopping device. There is a full range available from 6 to 240 tons, single and double eccentric.

PLEASE LET US DEMONSTRATE TO YOU

VAUGHAN
ASSOCIATES LIMITED

4 QUEEN STREET, CURZON STREET, LONDON, W.1
Telephone: GROSVENOR 8362-5

Midland Office: WILFORD CRESCENT, NOTTINGHAM Telephone: NOTTINGHAM 88008

Nelson stands on his column

Intal stands on its own good name

INTAL

THE INTERNATIONAL TWIST DRILL CO. LTD.

INTAL WORKS · WATERY ST. · SHEFFIELD · 3

Telephone: 23072-3 Telegrams: Fluted, Sheffield

OBTAINABLE FROM YOUR STOCKIST

London Stocks: 16, ALDERSGATE ST., E.C.1.

Phone: MON. 3505

Glasgow Stocks: 50, WELLINGTON ST., C.2.

Phone: CITy 6994

FOR REALLY RUGGED WORK YOU NEED

Don

SINTERED METAL FRICTION MATERIALS

. . . second to none for:—

* Operative smoothness
* Frictional stability
* Long working life

Details and information from:—

SMALL & PARKES LTD

HENDHAM VALE WORKS, MANCHESTER 9

COLlyhurst 2511

LONDON OFFICE 76 VICTORIA ST., SW1 VICtoria 1845/6

Adjustable TENSION WRENCHES

NEW NUT TIGHTENING TORQUE DATA Leaflet available

SPEED WITH ACCURACY

STURDY & RELIABLE.
COMPACT DESIGN.
ADJUSTABLE.
3 PHASE SIGNAL BY SIGHT, SOUND & FEEL.
LOW PRICE.

6 Models cover 2-400 foot lbs for $\frac{3}{8}''$ $\frac{1}{2}''$ and $\frac{3}{4}''$ Sq. Drive

TOOL UP WITH

CHROME ALLOY STEEL BRITOOL SERVICE TOOLS

STAND No. 520, TRANSPORT SERVICE SECTION, MOTOR EXHIBITION, EARLS COURT.

JENKS BROS. LIMITED,

BRITOOL WORKS, BUSHBURY, WOLVERHAMPTON, STAFFS.

The finest all round degreasant and cleaner

FOR ALL METAL PARTS, MACHINERY, FLOORS, WALLS, PAINTWORK, TILES, ETC.

Grease and oily deposits don't stand a chance when GUNK gets to work! Simply brush or spray GUNK on wherever is to be cleaned—rinse off with water—and see how all traces of grease and oil are instantly removed. So simple to use and so effective! There is no bothersome heating or mixing either. GUNK is supplied ready for use in 5, 10 and 40/45 gallon drums.

Send today for a free trial tin and see for yourself the remarkable cleaning powers of GUNK.

Manufactured by BENNETT (HYDE) Ltd. · Boston Mills · Hyde · Cheshire Tel. Hyde 1377-8-9

H. A. RIEDL LTD.

take pleasure in announcing, as from OCTOBER 16th, an important addition to the range of

HAREX

QUALITY PRODUCTS

TIE ROD ENDS & COMPLETE TIE ROD ASSEMBLIES
DRAG LINK ENDS & COMPLETE DRAG LINK ASSEMBLIES
BALL JOINT ASSEMBLIES • KING BOLT SETS
FRONT SUSPENSION KITS
SPRING SHACKLE ASSEMBLIES, BOLTS & NUTS

For Export only — New Illustrated Catalogue available on request

All Enquiries to:
29 STATION ROAD, NEW BARNET, HERTS, ENGLAND
Telephone: BARNET 5001 (3 lines) *Telex:* HAREXCO, BARNET

HAREX *Quality Products*

OTHER HAREX PRODUCTS INCLUDE:

PISTONS
•
CYLINDER LINERS
•
BEARINGS
•
WATER PUMPS
•
WATER PUMP REPAIR KITS
•
BELLOWS TYPE OIL SEALS
•

CLASSIFIED ADVERTISEMENTS

RATE 4d. per word, minimum 4/-. Each paragraph charged separately. Box number 5 words—plus 1/-. Advertisements for the November, 1957, issue should be to hand not later than first post 22nd October. No responsibility accepted for errors.

MACHINERY FOR SALE

FOUR Magec No. 10 Automatic Wiring and Edging Machines for sale. Capacity—minimum, 24 S.W.G. sheet and 10 S.W.G. wire; maximum, 14 S.W.G. sheet and ¼" wire, or 16 S.W.G. sheet and $\frac{3}{16}$" wire. Two rates of feed, 6 ft. and 18 ft. per min. Treadle clutch. Motor drive for 400/3/50. Photo—F. J. Edwards Limited, 359 Euston Road, London, N.W.1. [5594

SERVICES OFFERED

DESIGN Office Capacity available for Mechanical and Production Engineering. Special Machines and Projects. Norris Brothers Ltd., 53 Victoria Street, London, S.W.1. Tel.: Abbey 5444. [5546

MOTOR Trade component manufacturers—prepare for European Free Trade. Eurosales Ltd. provides you with European representation. Large established connections available with Continental Automobile and Machine Tool Manufacturers. For details write to Eurosales Ltd., 16 Selly Wick Road, Birmingham, 29. [5595

TUITION

A.M.I.M.I., City and Guilds, A.M.I.Mech.E., etc., on "No Pass-No Fee" terms. Over 95% successes. For details of Exams and Courses in all branches of Auto., Aero, Mechanical Eng., etc., write for 144 page handbook—Free. B.I.E.T. (Dept. 643), 29 Wright's Lane, London, W.8. [5553

SITUATIONS WANTED

CHIEF Engineer (Design) with large Automobile Manufacturers, desires change. Age 40. Would consider position as Technical Representative if satisfactorily remunerative. Box 1952, c/o Automobile Engineer. [5596

SITUATIONS VACANT

APPLICATIONS are invited for pensionable posts as Examiners in the Patent Office to undertake the official scientific, technical and legal work in connection with Patent applications.

AGE at least 21 and under 35 years on 1st January, 1957, with extension for regular Forces' service.

CANDIDATES must have (or obtain in 1957) 1st or 2nd Class Honours in Physics, Organic or Inorganic Chemistry, Mechanical or Electrical Engineering or in Mathematics, or an equivalent qualification, or have achieved a professional qualification, e.g. A.M.I.C.E., A.M.I.Mech.E., A.M.I.E.E., A.R.I.C For a limited number of vacancies, candidates with 1st or 2nd Class Honours degrees in other subjects—scientific or otherwise—will be considered. Exceptionally, candidates otherwise qualified by high professional attainments will be considered.

STARTING pay for 5-day week of 42 hours in London between £605 and £1,120 (men), according to post-graduate (or equivalent) experience and National Service. Maximum of scale £1,345. This salary scale is being increased by approximately 5 per cent. Women's pay above £605 slightly lower but is being raised to reach equality with men's in 1961. Good prospects of promotion to Senior Examiner, rising to £2,000 (under review) and reasonable expectation of further promotion to Principal Examiner.

APPLICATION form and further particulars from Civil Service Commission, Scientific Branch, 30 Old Burlington Street, London, W.1, quoting S 128/57 and stating date of birth.

INTERVIEW Boards will sit at intervals, as required. Early application is advised. [5592

DEVELOPMENT Engineer required to work on the development of rotating A.C. and D.C. low voltage machines and their associated control gear.

THIS work is of an interesting and responsible nature in connection with important Ministry contracts, and affords good opportunities to gain useful experience and for promotion to senior status.

MINIMUM qualifications required are H.N.C. or degree. Salary within the range £750 to £1,050 p.a.

APPLICATIONS in writing to Box 1832, c/o Automobile Engineer, Ref. D.E. [5593

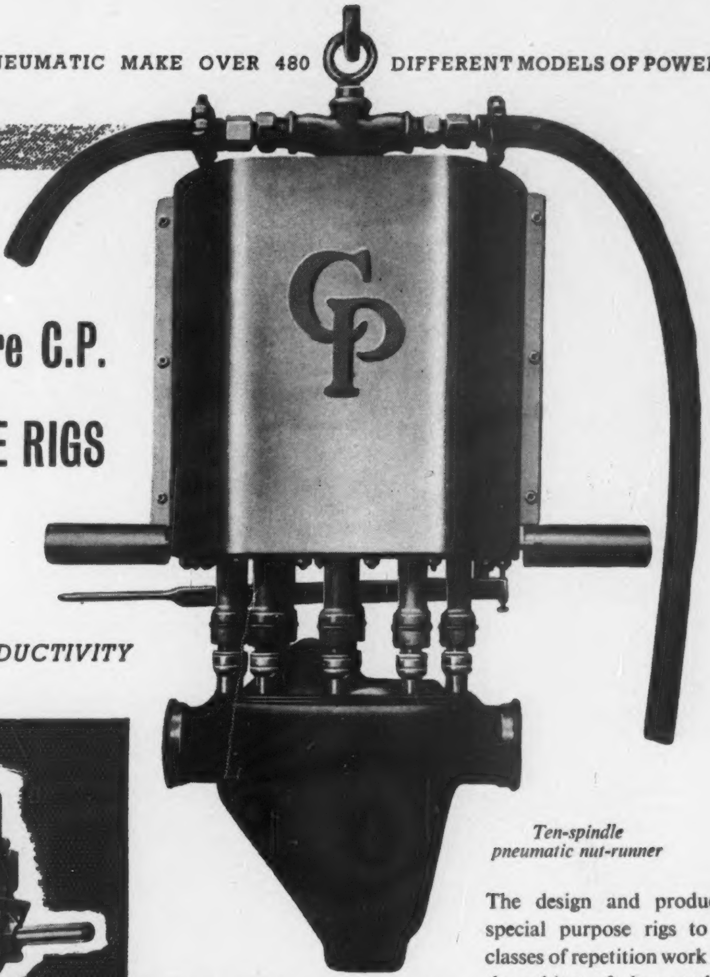
INDEX TO ADVERTISEMENTS

Printed in Great Britain for the Publishers, ILIFFE & SONS LTD., Dorset House, Stamford Street, London, S.E.1, by James Cond Ltd., Charlotte Street, Birmingham 3. "Automobile Engineer" can be obtained abroad from the following: AUSTRALIA & NEW ZEALAND: Gordon & Gotch Ltd. INDIA: A. H. Wheeler & Co. CANADA: The Wm. Dawson Subscription Service Ltd.; Gordon & Gotch Ltd. SOUTH AFRICA: Central News Agency Ltd.; Wm. Dawson & Sons (S.A.) Ltd. UNITED STATES: Eastern News Co. Entered as Second Class Matter at the New York, U.S.A., Post Office,

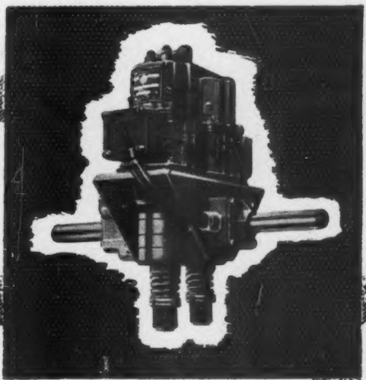
CONSOLIDATED PNEUMATIC MAKE OVER 480 DIFFERENT MODELS OF POWER TOOLS

These are C.P. MULTI-SPINDLE RIGS

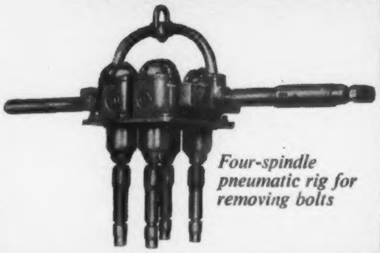
FOR INCREASED PRODUCTIVITY



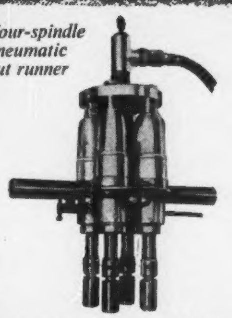
Ten-spindle pneumatic nut-runner



Hicycle electric two-spindle nut runner



Four-spindle pneumatic rig for removing bolts



Four-spindle pneumatic nut runner

The design and production of special purpose rigs to suit all classes of repetition work has been the subject of close study by CP development engineers, and a great variety of these units has been supplied to industry. Standard CP pneumatic stall torque motors or Hicycle electric motors are incorporated in these special rigs for setting all sizes of nuts, bolts and screws. In each case the units can be set to a predetermined torque figure. Each job requires separate consideration, but the illustrations here are typical examples of the types of units which can be produced. Write for catalogue SP 437.

Consolidated Pneumatic

CONSOLIDATED PNEUMATIC TOOL CO. LTD., 232 DAWES ROAD, LONDON, S.W.6

AIR COMPRESSORS · PNEUMATIC TOOLS · ELECTRIC TOOLS · CONTRACTORS EQUIPMENT · ROCK DRILLS · PUMPS · DIAMOND DRILLS

CP188